HEAT BALANCE OF A HVID ENGINE

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Heat Balance of a Hvid Engine

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Professor of Mechanical Engineering
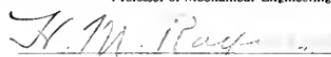
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The Heat Balance of a Hvid Engine

HEAT BALANCE OF A HVID ENGINE

AUTHOR'S NOTE

It is not our purpose to offer a complete analysis of the Hvid engine. Rather, the work we have done is merely a beginning, and our time has been largely spent in putting the engine in condition to test. This work is described in detail in the text.

We wish to thank Professor Roesch for the opportunity he has given us to work on our own initiative, and for his constant aid and advice.

Professor Perry has also lent us his aid in the shaking-forces analysis.

We are further indebted to Mr. Larson of the class of 1917, who has done some special work on the characteristics of different fuels in the injector. This work will be presented by Mr. Larson personally.

<div style="text-align:right">

L. E. Jones

E. C. Schwachtgen

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Heat Balance of a Hvid Engine

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HEAT BALANCE OF A HVID ENGINE

THE HVID ENGINE

The Hvid Engine is of necessity a heavily
built machine and is designed primarily for
heavy duty work. It is built in single and
multi-cylinder models, and is used largely for
farm and machine shop work. A four cylinder
25 horse-power engine is built by the Common-
wealth Motors Co. of Joliet, Ill., for marine
work. The price of this engine (January 1920)
is $2000. This model should prove popular,
despite the high first cost (which by the way,
is as cheap or cheaper than any other <u>success-</u>
<u>ful</u> kerosene engine adapted to this work) be-
cause of its reliability and inherent economy.

The engine used (Fig. 1) is built by Sears
Roebuck & Co. of Chicago, under the trade name
"Thermoil". It is of the single cylinder
horizontal type and developes 8 horse power.
The price is $300. Early models of this engine
gave trouble, due to incomplete development of

Cross Section of Cylinder and Cylinder Head to Show Operation of Engine

FUEL RESERVOIR

SPEED CONTROL WHEEL

FUEL VALVE LEVER

FUEL PIPE

GOVERNOR ROD LEVER

FUEL VALVE ROD LEVER

GOVERNOR ROD

EIGHT RINGS TWO IN EACH GROOVE

FUEL MEASURING PIN

FUEL INLET VALVE

FUEL VALVE ROD

FUEL CUP

GROUND PISTON AND CYLINDER

LOWER FUEL VALVE ROD LEVER

AIR INLET VALVE

FUEL VALVE ADJUSTING SCREW
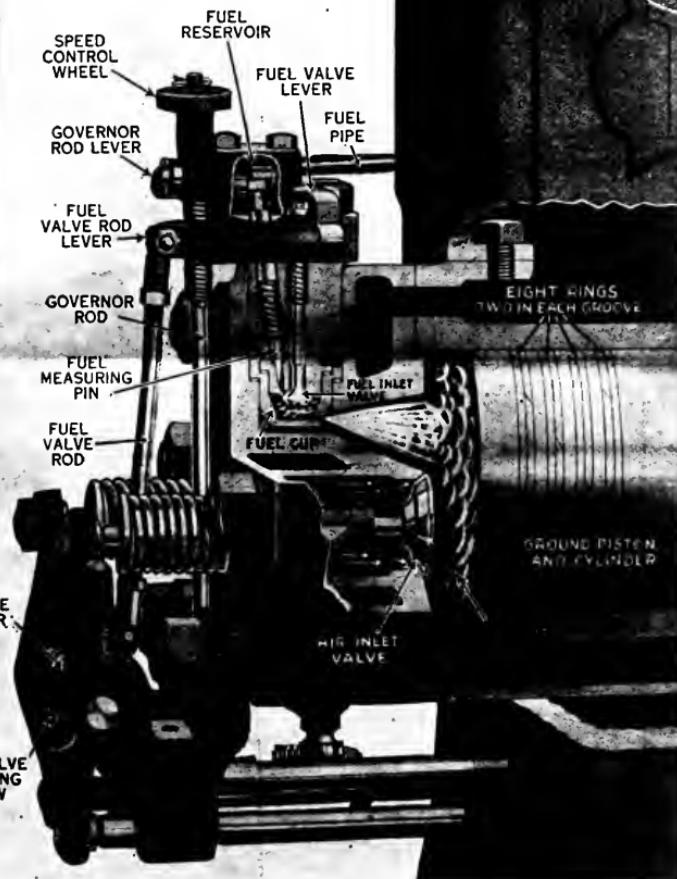
Figure 5

the injector, but this obstacle has been overcome and the present models are considered as reliable as the ordinary farm engine.

The engine is governed by a high speed centrifugal governor,geared to the timing gears. Governoring is effected by reduction of fuel, and a full-load speed of 450 r.p.m. is maintained. (Hvid engines have been built that operate successfully at 1500 r.p.m.).

The inherent advantages of this type of engine are:

1. Mechanical Simplicity
2. Low fuel consumption at all loads
3. Absence of electrical or hot bulb ignition
4. Ability to start cold on any free-flowing fuel
5. Absence of carburetting device
6. Low water jacket losses
7. Ease of lubrication
8. Lack of carbon troubles
9. Constant compression
10. Good torque characteristics
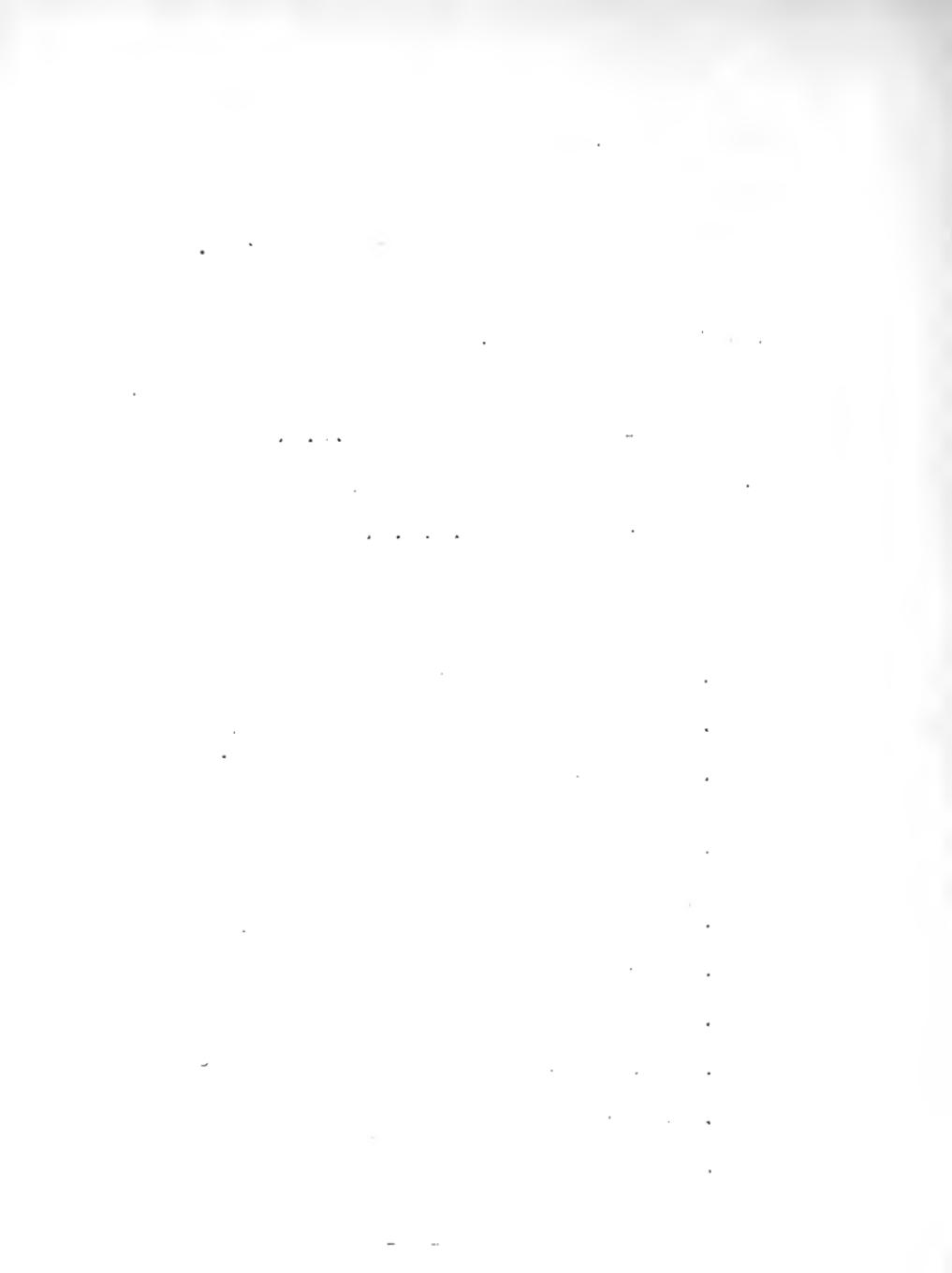
EXPLANATION OF CYCLE

The basic four cycle (Otto cycle) principle is used, with this modification, pure air, instead of a mixture of air and gaseous fuel, is drawn in the cylinder on the suction stroke. Two values are employed, actuated as in the usual gasolene engine by a camshaft geared from the crankshaft to run at one-half engine speed.

Ignition is effected by the compression temperature (900 to 1000 deg. fahr. due to a compression pressure of 425 to 475 lbs. per sq. in.). Referring to figure 2, the phases of the cycle are as follows:

SUCTION STROKE: As the piston moves down the stroke (toward the crankshaft), the inlet valve "A" opens and a charge of pure air is drawn into the cylinder. The fuel valve "B", (actuated by the push rod "R") opens simultaneously with the inlet valve, and fuel enters the cup "C", due partly to the action of gravity and partly to inhalation. The amount of fuel admitted is regulated by the metering -
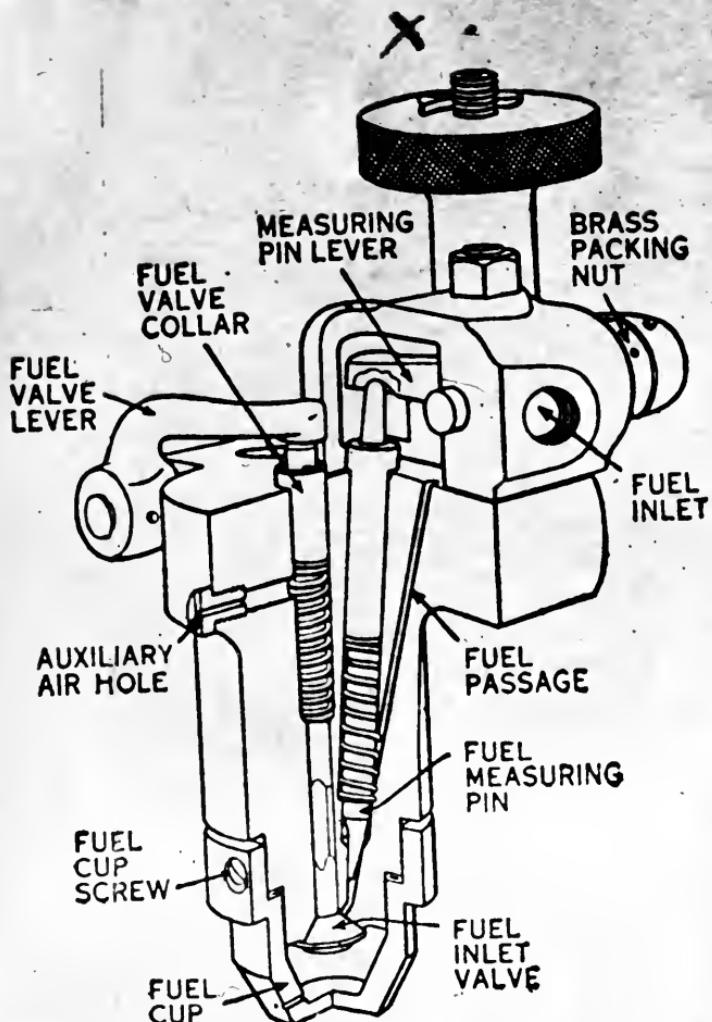
Figure 6.

pin "E", which is linked to the governor. As
the fuel enters the cup, a small charge of air
is drawn through the hole "F" and enters the cup
through the guide "G". As the piston nears the
end of the stroke, the inlet and fuel valves are
closed.

COMPRESSION STROKE: On this stroke all the
valves are closed and the entrapped air is com-
pressed to a pressure of 425 to 475 lbs. per sq.
in. As the pressure builds up, air is forced
into the cup through the holes "H" and a pressure
nearly equal to that in the cylinder is finally
attained. At the temperature due to this pres-
sure, the more volatile parts of the fuel ignite
spontaneously, and a "primary" explosion takes
place within the cup. The force of this explosion
drives the heavier constituents of the fuel thru
the holes into the cylinder. At this stage the
piston has reached the end of the stroke and the
compression is at a maximum, and the temperature
is high enough to ignite the fuel. (The fuel

burns, rather than explodes). Due to air defi-
ciency, the amount of fuel burned in the cup is
infinitesimal.

POWER STROKE: As the fuel is forced into
the cylinder it is thoroughly atomized, and is
ignited by the temperature due to compression.
The consequent expansion forces out the piston,
as in the ordinary gas engine.

EXHAUST STROKE: As in the ordinary four
cycle engine, the exhaust valve opens at the end
of the power stroke and the products of combus-
tion are expelled by the piston on the return
stroke.

FORCE FEED OILER AND FUEL PUMP

DROP FORGED CONNECTING ROD WITH REMOVABLE DIE CAST HIGH PRESSURE BABBITT BEARING

MAIN BEARINGS ARE OF REMOVABLE DIE CAST HIGH PRESSURE BABBITT

LARGE HEAVY DROP FORGED CRANK SHAFT

ONE PIECE CAM SHAFT OPERATING VALVE RODS REGULAR AUTOMOBILE CONSTRUCTION

GROUND PISTON AND CYLINDER WITH EIGHT RINGS TWO IN EACH GROOVE
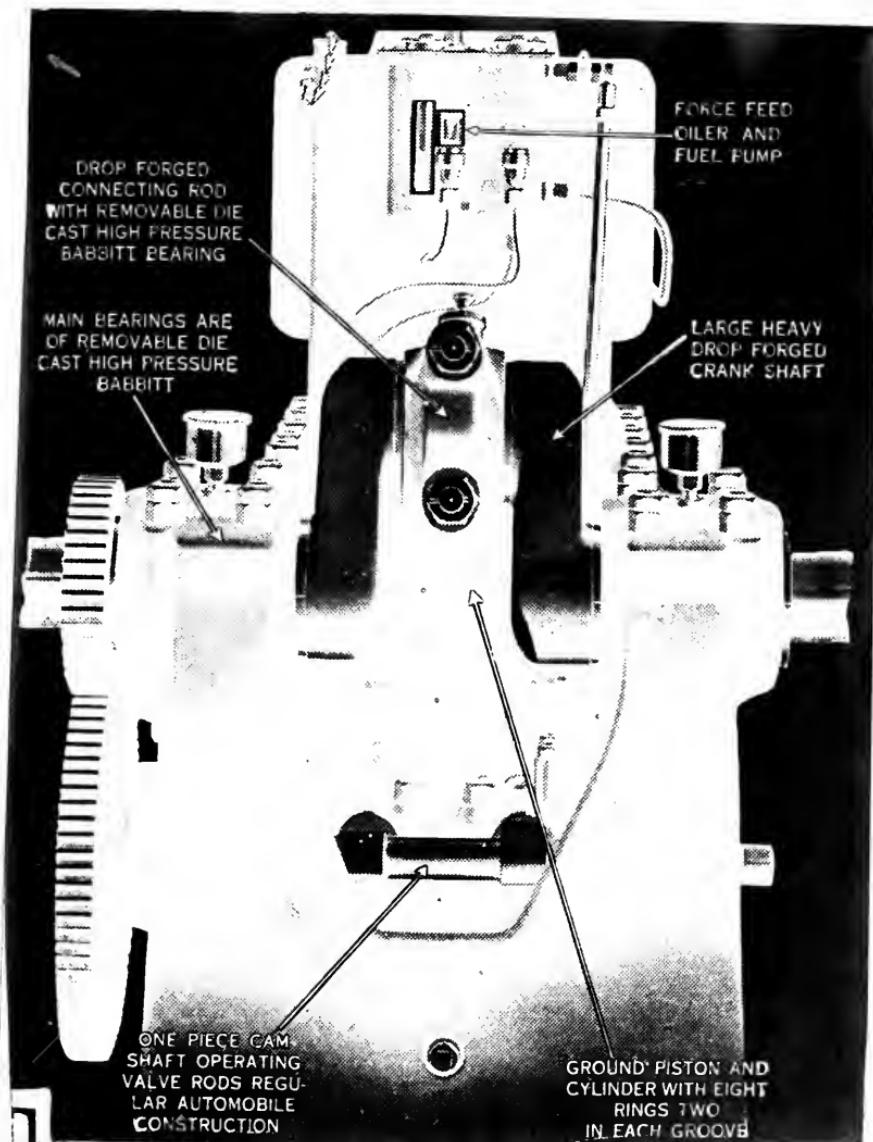
OPERATION

SPEED CONTROL: The speed control wheel
(Fig. 1) regulates the amount the fuel value
opens, and thus regulates the amount of fuel
fed per stroke to the fuel cup. When the wheel
is screwed up as far as it will go, the maximum
value opening is allowed, and the engine runs at
normal speed. Screwing the wheel down reduces
the supply of fuel and hence reduces the speed.

A noticeable amount of play is observed
when the wheel is screwed up. If the wheel is
screwed down until this play is eliminated, the
fuel supply is entirely shut off. (In order to
cut off the supply of fuel quickly, in case of
an accident, we inserted a stop-cock in the fuel
line, close to the injector).

LUBRICATOR AND FUEL PUMP: The pressed steel
box attached to the back of the water hopper
(Fig. 3) contains a combination lubricator and
fuel pump. The box is divided into two sections,
one an oil reservoir and one a fuel reservoir.
Oil is pumped from the oil reservoir to the pis-

ton, and fuel is pumped from a tank in the base to the fuel reservoir. (The oil reservoir is filled by hand). From the fuel reservoir, the fuel flows under the action of gravity to the injector. The pumps are actuated by the camshaft and the link, as shown in Fig. 3.

TO START THE ENGINE: Before starting the engine, the oil and fuel reservoirs must be filled, and all grease cups should be screwed down. To quote from the instruction book furnished by the manufacturers:

"Release the compression in the cylinder by pulling the compression release lever up to the starting position (Fig. 4).

Turn on fuel by screwing up the speed control wheel.

With the starting crank, turn the flywheels rapidly through eight or ten revolutions. Remove the crank rapidly and immediately push the compression release lever down to the running position. The momentum of the flywheels will turn

SPEED
CONTROL
WHEEL

STARTING
POSITION

RUNNING
POSITION

COMPRESSION
RELIEF LEVER
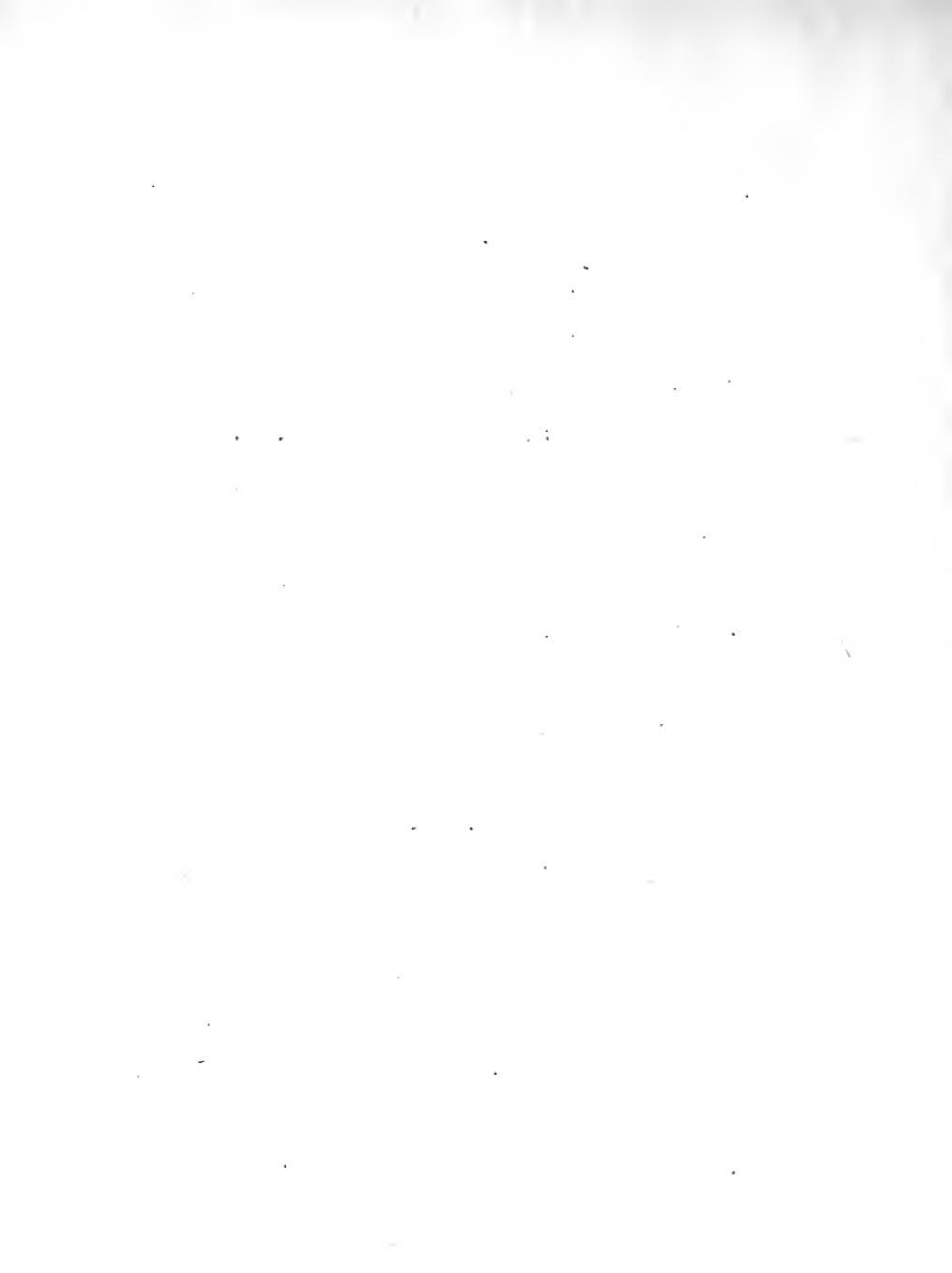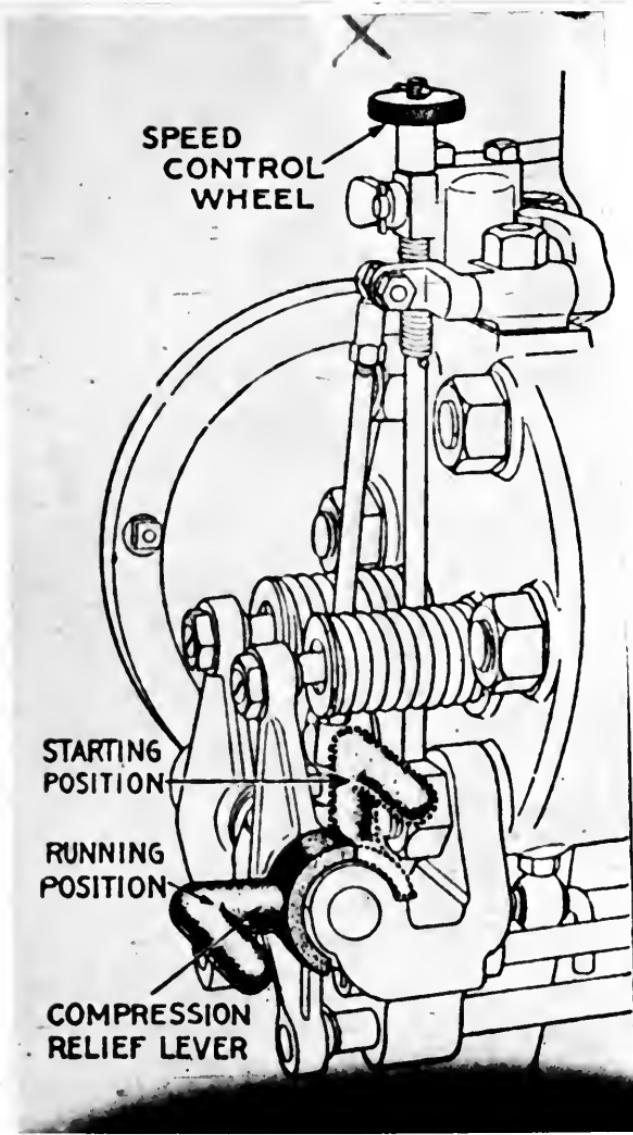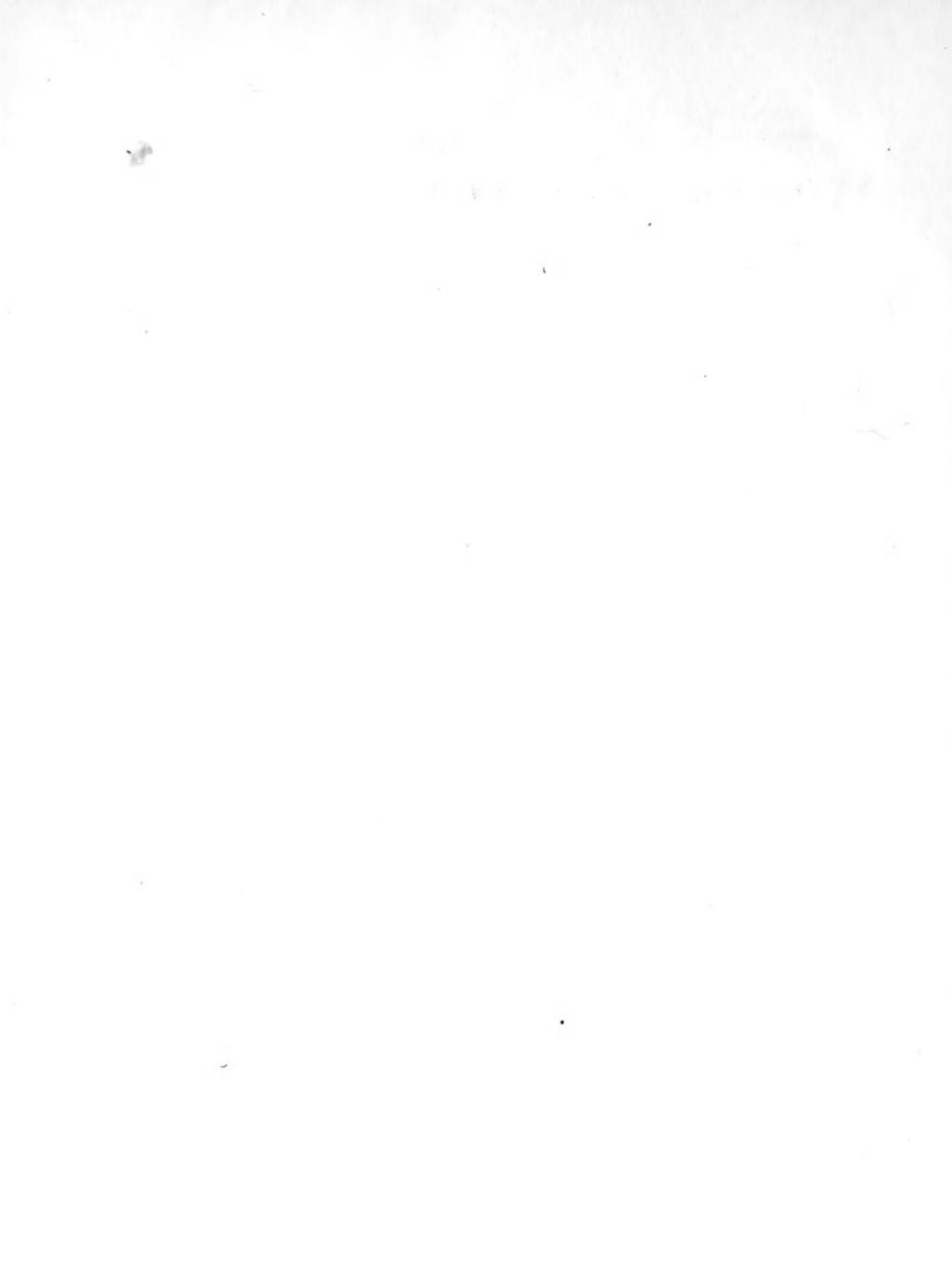
the engine over compression and the engine will start.

To shut down the engine, screw down the speed control wheel until the fuel is shut off."

The lubricator should be adjusted to give about 15 drops of oil per minute.

If the engine is to be started warm (i.e. after it has been in operation, and has not cooled off) less fuel will be required. Therefore the speed control wheel should be only partially screwed up. Indeed, it is possible to "flood" the engine by admitting too much fuel, in which case difficulty will be encountered in starting.

If the engine pounds upon starting, an excess of fuel is indicated, and the entire supply should be cut off until this excess is used. When the engine begins to slow down, the wheel should be screwed up until proper running conditions are reached.

FUEL INLET VALVE: The fuel inlet valve sometimes leaks, due to the formation of carbon, or

to dirt. When this condition occurs, the valve
must be ground in. To quote further from the
instruction booklet:

"To grind the fuel inlet valve, remove the
fuel injector from the cylinder head. Take off
the fuel cup and remove valve collar and valve.
Place a minute quantity of powdered pummice and
oil on the valve seat, and turn the valve slow-
ly back and forth on the seat. A gentle pres-
sure should be used. After grinding, the parts
must be washed in kerosene."

In placing the indicator on the cylinder
head, the rods which actuated the fuel valve had
to be replaced with special rods, made to clear
the indicator. As these rods were of different
length from the original ones, the timing of the
valve was changed. Resetting was accomplished
with the crank straight down (180 degrees from
the position shown in Fig. 3) and the compression
release lever in the running position. At this
phase, there is supposed to be between .005 and

.01 of an inch clearance between the valve
tappet and valve stem. This clearance is ad-
justed by the fuel valve adjusting screw shown
in Fig. 1.

Work Preliminary to Testing

MOUNTING THE ENGINE: We received the
engine bolted on skids. These were left in
place and a bolt was sunk in the concrete floor
at each end of either skid. (Special bolts,
having a nut on each end, were used. Holes were
bored in the skids first, and then the relative
positions of these holes were laid off on the
concrete floor. Holes were drilled with a star
drill and a nut with a bolt threaded in place
was inserted in each hole. Melted lead was
then poured around each bolt. When the engine
is removed, the bolts can be readily withdrawn
from the nuts, leaving them in the floor for
future use.

The engine was run unloaded after being bolt-
ed in place, but excessive vibration resulted.
Accordingly, two more holes were placed in each
skid, and bolts to correspond were sunk in the
floor. A second test proved the vibration to be
satisfactorily reduced.

For a permanent setting the engine should be

mounted directly on a concrete base.

THE REDUCING MOTION

To indicate the engine a special head with
a hole tapped in it to receive the indicator
was procured from the manufacturers. The reduc-
ing motion, however, had to be made in the shop.

A reducing motion must reproduce, point by
point, the motion of the piston. When the en-
gine has a cross-head, the familiar pantograph
motion is usually employed, and the length of
card produced is about three inches.

As this engine has a full-trunk type pis-
ton (customary in gasolene and small gas engine
practice) this method could not be used. Our
apparatus consisted of a small eccentric, four
and one fourth inches in diameter, having a
throw of five eights of an inch. This gave an
indicator card one and one fourth inches in
length. The parts were made small, in an at-
tempt to eliminate inertia forces and the re-
sulting discrepancies.

The eccentric circle was keyed to the crank-
shaft at zero degrees with the crank. A push
rod extended along the center line of the engine,
close to the indicator. Three bearings support-
ed the rod, which was made of steel, three quar-
ters by three eighths of an inch. To reduce
weight, the rod was melted out to an I beam
section, except at the bearings.

The rod must be placed precisely on the
center line of the engine. If it is displaced
either way, the motion of the piston will not
be accurately reproduced, due to a difference
in angularity on the forward and return strokes.
(This fact is made use of in shaper practice,
where a "quick return" motion is realized by
the use of an off-set rod).

The eccentric was placed on the right hand
side of the engine, inside the flywheel, and
set the flywheel out about seven-eighths of an
inch. This seriously interferred with cranking
the engine, as room enough was not left to fit
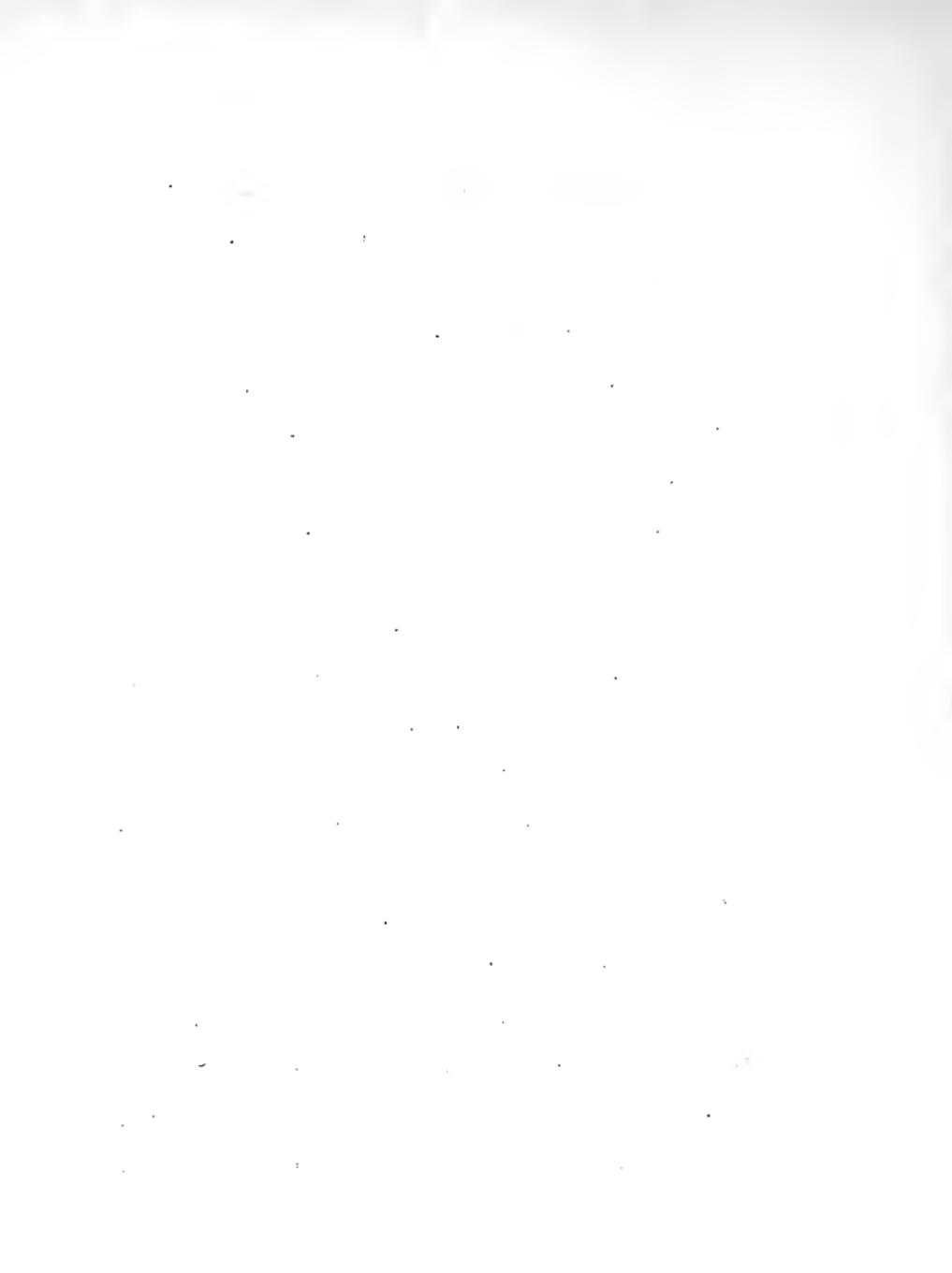
the crank handle properly on the shaft.

Unfortunately, this objection was removed by the failure of the apparatus. After ten minutes of running, the eccentric strap failed, wrecking the entire reducing motion. Whether the failure was due to a seized bearing, or an inherent weakness in the strap, has not been determined. However, it is probable that the strap gave away, as the inertia forces at 450 r.p.m. are large.

In redesigning the apparatus, more rugged construction was used. (See blue print of eccentric appended). The throw was increased to one inch, giving a two inch card. To obviate the difficulty in cranking, the new eccentric was placed on the other side of the engine.

The push-rod used was made of five-eighths cold rolled steel shaft of circular cross-section. Two bearings were used, in place of three, as the larger rod was less subject to flexure. We found by experience that the

simplest way of lining up such a shaft was by babbitting the bearings with the shaft lined up in place. (Previous to the babbitting, a steel bearing was used. This bearing seized, but due to the rugged construction employed, little damage was done, beyond bending the rod).

CONSTRUCTION OF PRONY BRAKE

A two-foot brake drum was made from plans
furnished by the Institute. The machine work
was done under Mr. Agle's supervision (by the
author's).

A large pulley runs "sweeter" than a small
one, due probably to the increased rim veloc-
ity. It was for this reason that a two foot
pulley was used.

A brake band of steel, lined with wood
blocks, was made as per the submitted blue
print. The brake arm (distance from the cen-
ter of the crankshaft to the point of applica-
tion of the load) was made 31.5 inches. Sub-
stituting this value in the formula

$$H.P. = \frac{2\pi R N W}{33000}$$

there results the equation

$$H.P. = \frac{N W}{2000}$$

where N = R.P.M.

W = net load on scale pan in lbs.
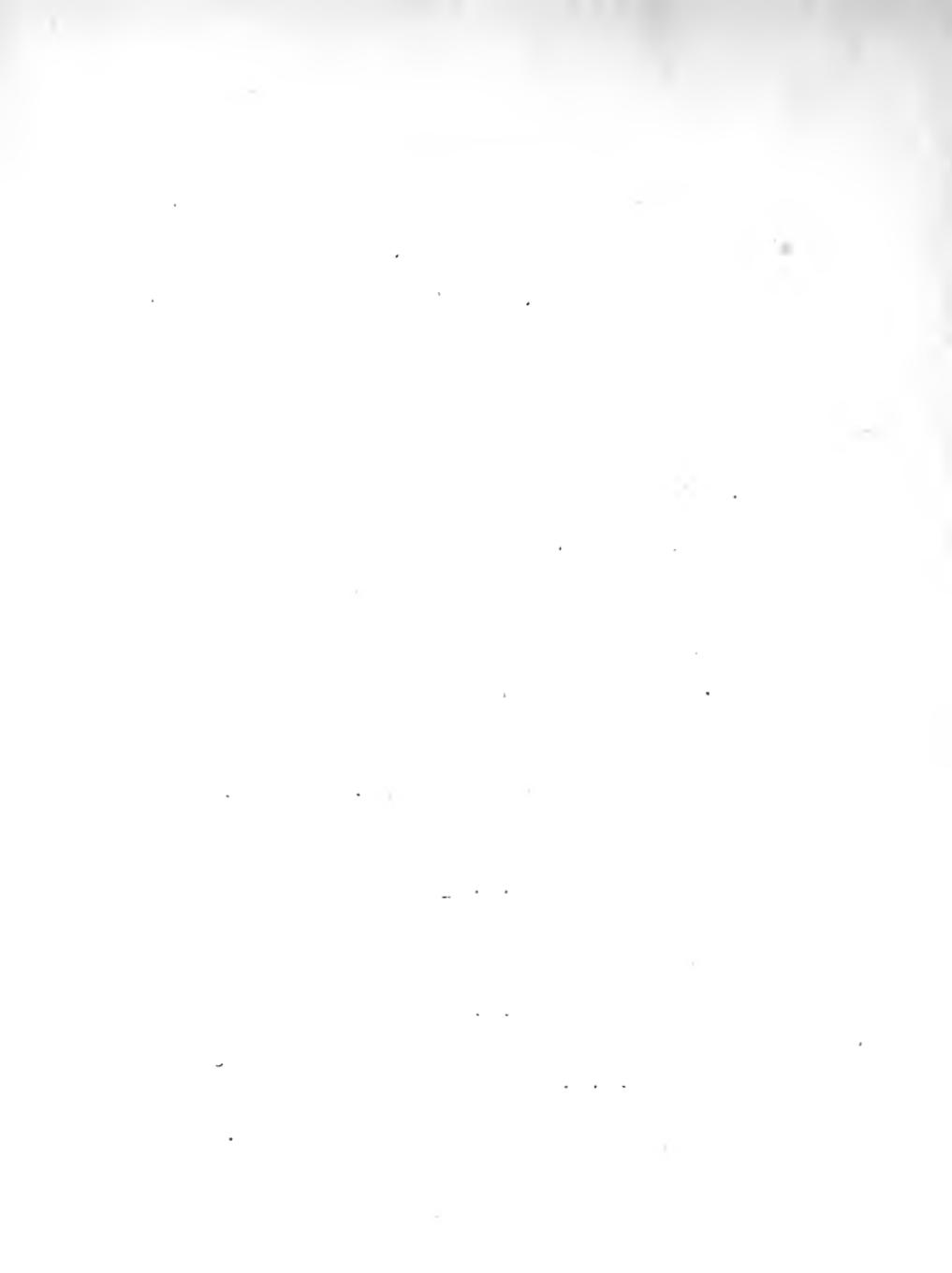
The Heat Balance

We did not have sufficient time to run a complete heat balance, but the engine has been left in condition to run this test.

Several runs for horse power were made,and a typical indicator card (Fig. 5) is included in this report.

At Professor Roesch's suggestion, we have included a heat balance, using data taken from an article in the January 1920 number of the Scientific American Monthly. This article was written by Mr. E. B. Blakely, who used data collected from a series of tests run by Prof. Roesch at the Institute.

Further information can be obtained by referring to the "Proceedings of the American Society of Mechanical Engineers" for December, 1919.

The heat balance gives the mechanical and thermal efficiencies for both brake and indicated horse power. The engine used by Prof. Roesch was the same size and model as the engine we

TYPICAL
INDICATOR CARD

PRESSURE - LBS PER SQ IN.

PER CENT OF STROKE
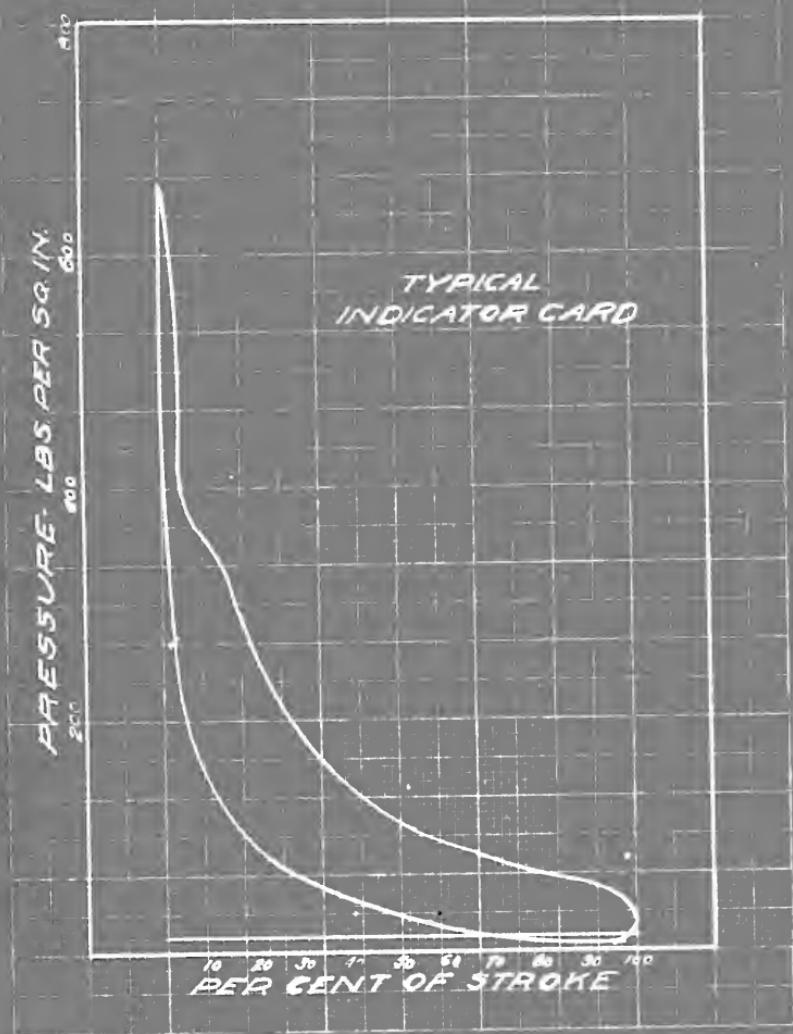
FIG. 7.

used (5-3/4 in. bore and 9-in. stroke).

For testing purposes, the engine was flex-
ibly connected to a Sprague electric dynamometer
by two Spicer universal joints.

Engine speeds and fuel weights were obtained
by electrically operated devices. The water
jacket losses were determined by weighing the
water used and recording the temperature dif-
ferences. The sensible heat of the exhaust
gases was determined by the calorimeter method.

Prof. Roesch also took indicator cards,
but (to quote from Mr. Blakely's article)" ...
because of the probable errors due to the high
pressures involved, and the comparatively high
speed of the engine, these cards were uséd mere-
ly to study the valve settings and general events
of the cycle."

The engine was operated under various loads
and speeds with various adjustments of fuel
supply, compression and cup design. The final
compression pressure was set at 390 pounds per

square inch.

Figures 6 and 7 illustrate by curves the results of these tests.

EFFICIENCIES
5¾×9 HVID ENGINE

MECH. EFFICIENCY.

THER. EFF. (I.HP.)

THER. EFF. (B.H.P.)

PER CENT

BRAKE HORSEPOWER.

FIG. 5.

RADIATION AND
INCOMPLETE COMBUSTION

SENSIBLE HEAT OF
EXHAUST GASES.

JACKET LOSSES

B.H.P.

FRICTION

PER CENT

BRAKE HORSEPOWER
HEAT BALANCE
5¾ × 9 H.W.D. ENGINE
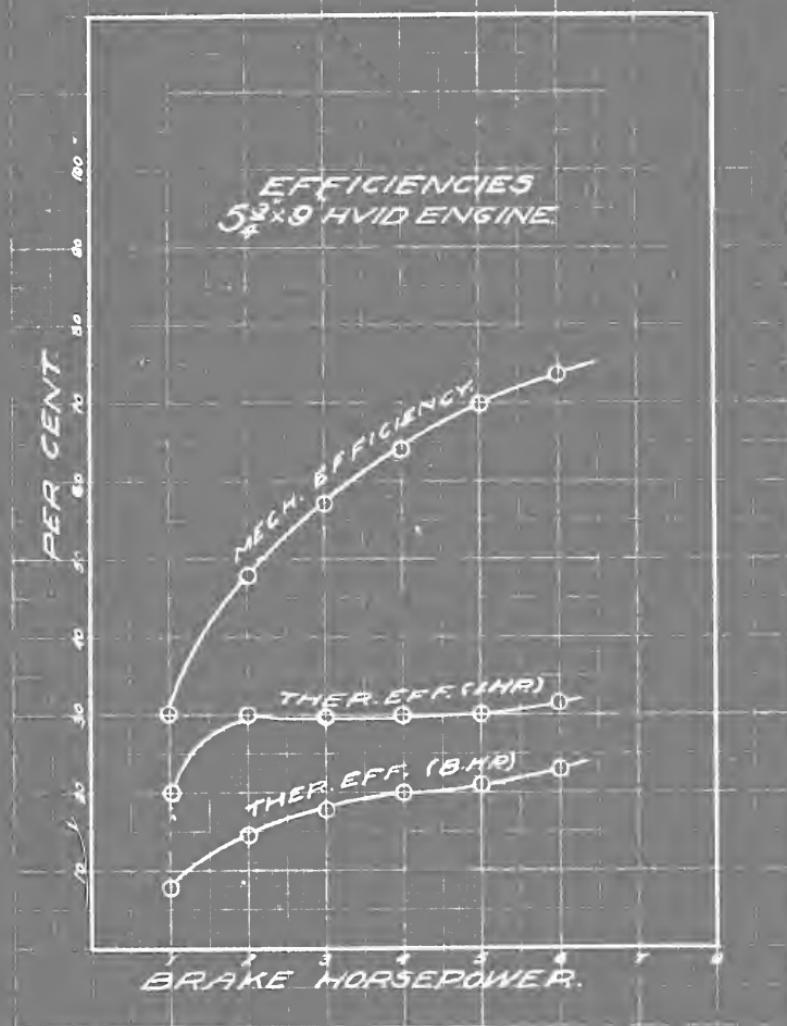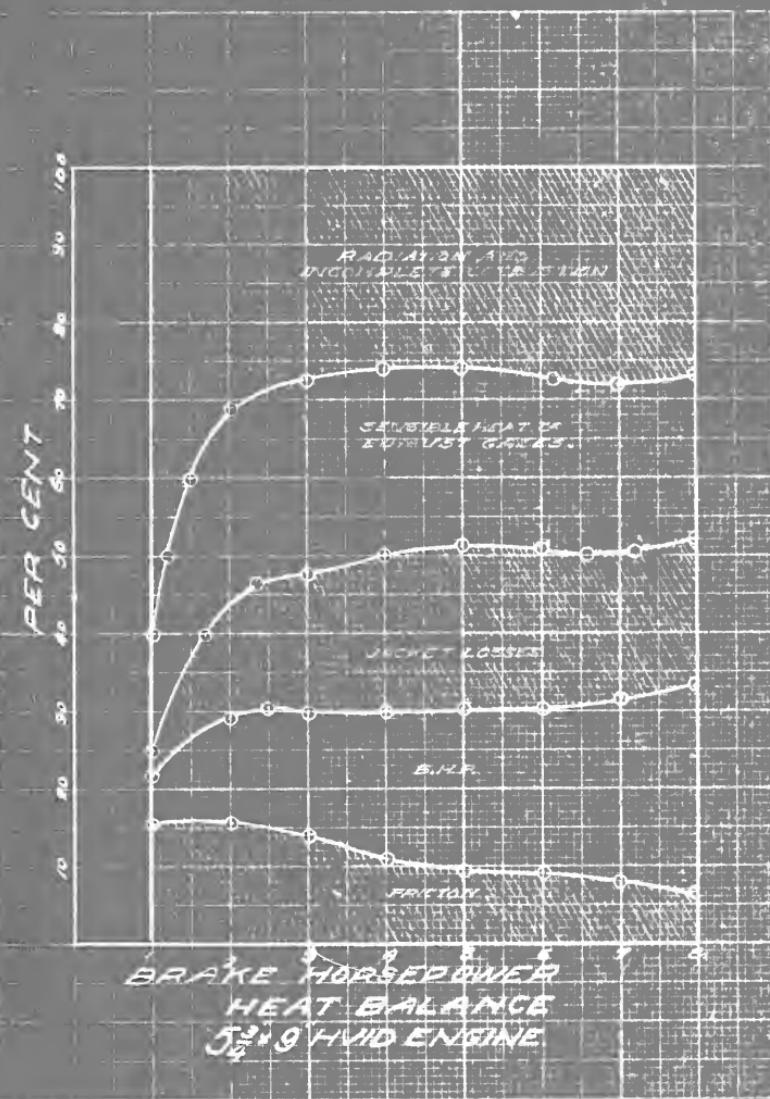
FIG. 6.
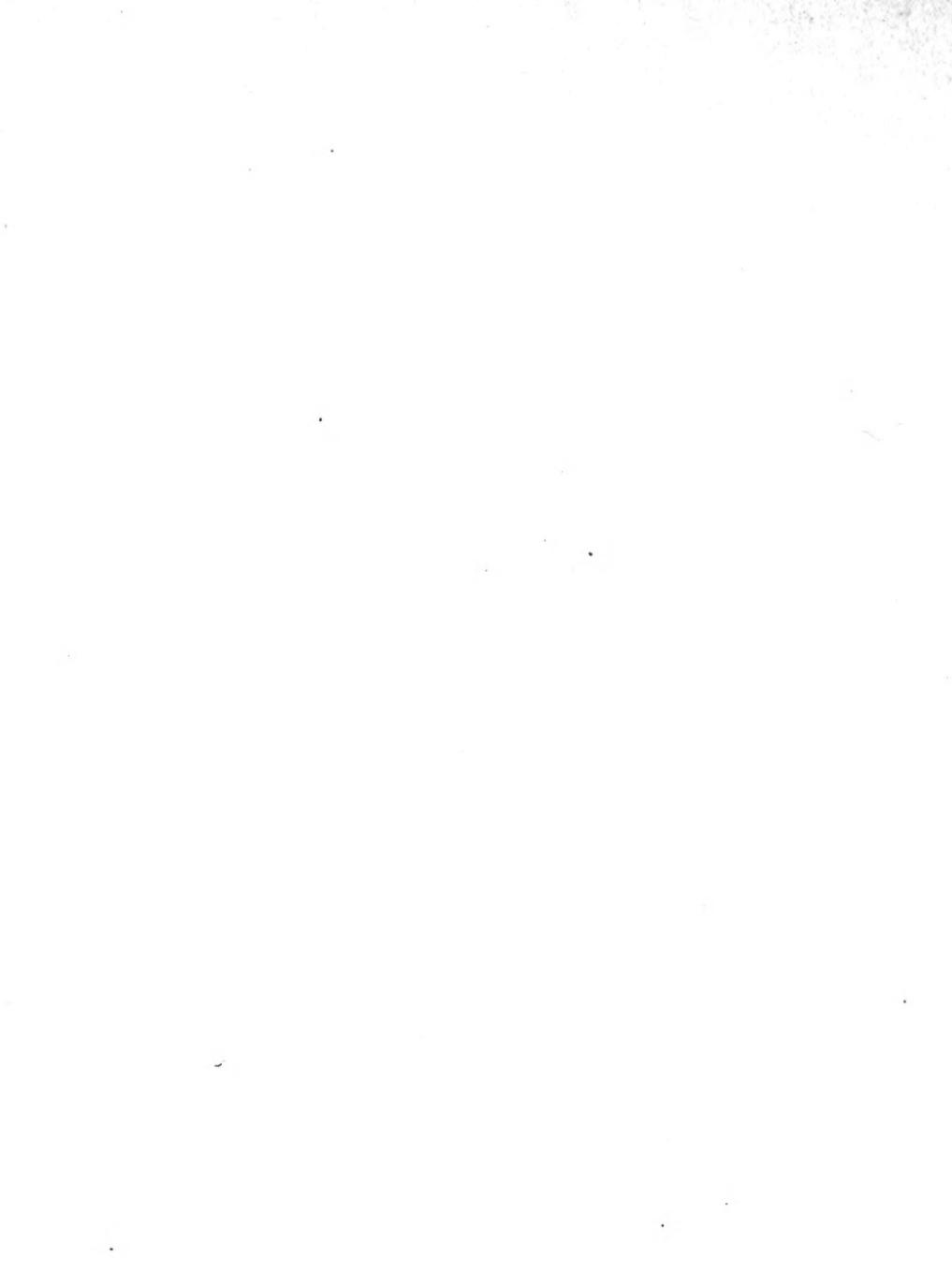
Shaking Force Analysis

DETERMINATION OF THE SHAKING FORCES

In addition to the heat balance proper a study of the shaking forces produced in the Hvid engine was made. Blue prints marked plates #1 and #2 illustrate diagrammatically the forces found and calculated in the following pages.

The pressure produced upon ignition of the kerosene vapor is expended in two ways. First, it accelerates the moving parts; second, it produces a pressure on the crank pin which may be resolved into two components, one of which produces rotation of the crank shaft, and the other pressure on the main bearings.

In order to get a clear idea of just how these forces act it is necessary to know something about the relative motion of the various moving parts. The piston has a simple motion of reciprocation, the crank an approximately uniform motion of rotation, while the connecting rod has a complex motion of combined oscillation and reciprocation. The reciprocating motion requires accelerating forces acting

alternately in opposite directions, as does also oscillation. On the other hand, a rotary motion induces a centrifugal force acting always toward the crank pin.

The various forces influencing rotation are illustrated in Fig. A, page   where

S = Net gas pressure

F = Translational inertia force of piston and parts

T = Translational inertia force of connecting rod

A = Angular inertia force of connecting rod

C = Centrifugal force of connecting rod

G = Gravity force due to the weight of the connecting rod

NET PRESSURE

The indicator card shown in plate 1 is the net pressure card, as the engine is single acting. The engine exhausts at 1-1/2 pounds above atmospheric pressure and the charge is drawn in at 1-1/2 pounds below atmospheric pressure.

FIG. A.
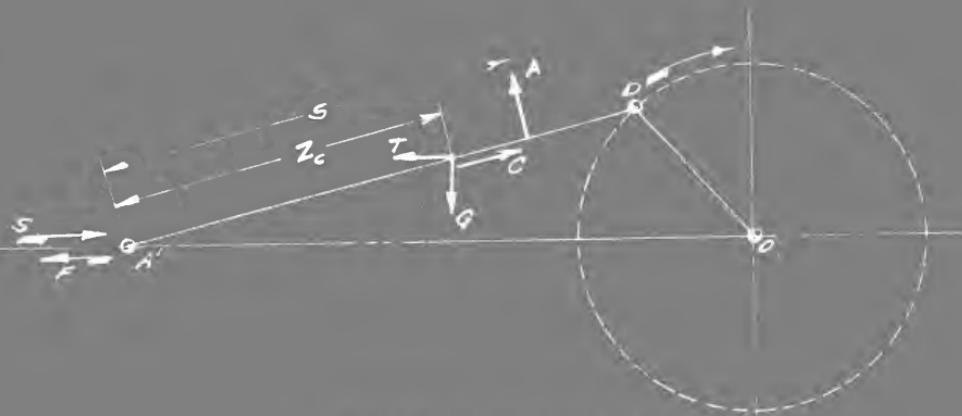
DIAGRAM OF FORCES

CONNECTING ROD.

At the end of each stroke the velocity of the piston and pin is zero, while near the middle of the stroke it is a maximum. To speed up these parts from rest to a maximum velocity and then to bring them back to rest again requires an alternately accelerating and retarding force.

This force F is found and calculated as follows:

Stroke = 9" .

r = 1/2 stroke = 9/2 = 4.5"

Piston diameter = 5.75"

Piston area = $1/4 \times 5.75^2$ = 29.959 sq.in.

Length of connecting rod = 22.5"

Ratio $r/L = \dfrac{4.5}{22.5}$ = 1/5

Weight of piston + pin = 25 pounds

Mass of piston + pin = $\dfrac{25}{32.2}$ = .777

N = revolutions per sec. = $\dfrac{450 \text{ r.p.m.}}{60}$ = 7.5

$F = -M \, \bar{w}^2 r \, (\cos \theta + \dfrac{r}{L} \cos \theta)$

$-M \, \bar{w}^2 \, r = - \dfrac{Mv^2}{2} \, r = - \dfrac{Mv^2}{r} = - M(2\pi r)^2 \, n^2$

$$- M \quad 2 n^2 \quad r$$

$$- 4 \quad ^2 - 39.478.$$

$$M N^2 R = \frac{.777 \times 56.25 \times .375}{25.959} = .632$$

$$F = - 39.478 \times .632 \times (\cos.\theta + 1/5 \cos. \quad \theta)$$

DETERMINATION OF (Cos. θ + 1/5 Cos. θ)

Angle	Function		Numerical Value
0°	(cos. 0° + 1/5 cos. 0°)	$(1 + 1/5 \times 1)$	+1.2
30°	(cos. 30° + 1/5 cos. 60°)	$(.866+1/5 \times .5)$	+ .966
60°	(cos. 60° + 1/5 cos. 120°)	$(.5 - 1/5 .5)$	+ .400
90°	(cos. 90° + 1/5 cos. 180°)	$(0 - 1/5 \times 1)$	- .200
120°	(cos. 120° + 1/5 cos. 240°)	$(-.5 - 1/5 \times .5)$	- .600
150°	(cos. 150° + 1/5 cos. 300°)	$(-.855 + 1/5 .5)$	- .766
180°	(cos. 180° + 1/5 cos. 360°)	$(-1 + .2)$	- .800

$$F = - 24.98 (\cos.\theta + 1/5 \cos \quad \theta)$$

Angle	(Cos.θ + 1/5 cos. θ)	Constant	F
0°	+ 1.2	- 24.98	- 29.60
30°	+ .966	"	- 24.05
60°	+ .400	"	- 9.97
90°	- .200	"	+ 4.98
120°	- .600	"	+ 14.98
150°	- .766	"	+ 19.13
180°	- .800	"	+ 19.95

TRANSLATIONAL INERTIA FORCE OF THE CONNECTING ROD

To produce the translation of the rod the

force T Fig. A, page   applied at the center of gravity is necessary. This force "T" is resolved into two forces, one "T" acting at the pin and the other "T_2" acting at the crank pin.

$T = T_1 + T_2$

$T_1 = - M_c \dfrac{(L-Z_c)}{L} \; W^2 N \; (\cos. \; \theta \; \dfrac{r}{L} \; \cos. \quad \theta)$

Weight of connecting rod $= 30$ pounds

M_c = mass of connecting rod $= \dfrac{30}{32.2} = .953.$

$Z_0 = 22.5 - 14.625 = 7.375$

$\dfrac{L - Z_0}{L} = \dfrac{7.375}{22.5} = .35$

$T_1 = \dfrac{(953. \; x \; .35 \; x \; .375 \; x \; 56.25)}{25.959} \; x \; constant$

$T_1 = .271$ x constant, or by ratio of masses,

$T_1 = \dfrac{30.5}{25} \; x \; .35 \; x \; F = .427 \; F.$

Angle	F	T_1 or .427 F
0°	− 29.60	− 12.63
30°	− 24.05	− 10.30
60°	− 9.97	− 4.25
90°	+ 4.98	+ 2.122
120°	+ 14.98	+ 6.40
150°	+ 19.25	+ 8.22
180°	+ 19.95	+ 8.53

$$T_2 = M_0 \frac{Z_0}{L} W^2 r \left(\cos \theta + \underline{} \cos \theta \right)$$

$$\frac{Z_0}{L} = \frac{14.625}{22.5} = .65$$

$$T_2 = \frac{(.953 \times .793 \times .375 \times 56.25) \times \text{constant}}{25.96}$$

or

$$T_2 = (1.22 \times 65 \times F = .793 F$$

Angle	F	T2 or .793 F
0°	- 29.60	- 23.43
30°	- 24.05	- 19.10
60°	- 9.97	- 7.91
90°	+ 4.98	+ 3.95
120°	+ 14.98	+ 11.89
150°	+ 19.13	+ 15.20
180°	+ 19.95	+ 15.83

ANGULAR INERTIA FORCE OF CONNECTING ROD

The angular inertia force "A" of the connecting rod when rotating about the wrist pin may be determined from the equation:

$$A = M_c Z_c W^2 \frac{r}{L} \sin \theta$$

$$- M_0 Z_c W^2 = \frac{.953 \times 4.625 \times 56.25}{12 \times 25.956}$$

$$x 4^2 = - 99.50.$$

Angle	sin θ	1/5 x sin θ
0°	0	0
30	.5	.1
60	.866	.173
90	.1	.20
120	.866	.173
150	.5	.1
180	0	0

Angle	1/5 sin θ	Constant	A
0°	+ 0	- 99.50	0
30°	+ .1	"	- 9.94
60°	+ .173	"	-17.21
90°	+ .2	"	-19.85
120°	+ .173	"	-17.21
150°	+ .1	"	- 9.94
180°	0	"	0

"A_1" and "A_2" are two forces acting at the wrist pin and crank pin respectively, which together give the same result as "A" acting at the center of percussion.

$$A_1 = \frac{A (L - S)}{L}$$

$$t = \frac{S}{g} = .31416 \quad \frac{S}{32.2} \qquad S = 3.26 \ t^2$$

10 suings in 7 seconds   .7 sec. = time of one suing.   .7 x 2 = 1.4 sec. = time of 1 period of connecting rod.
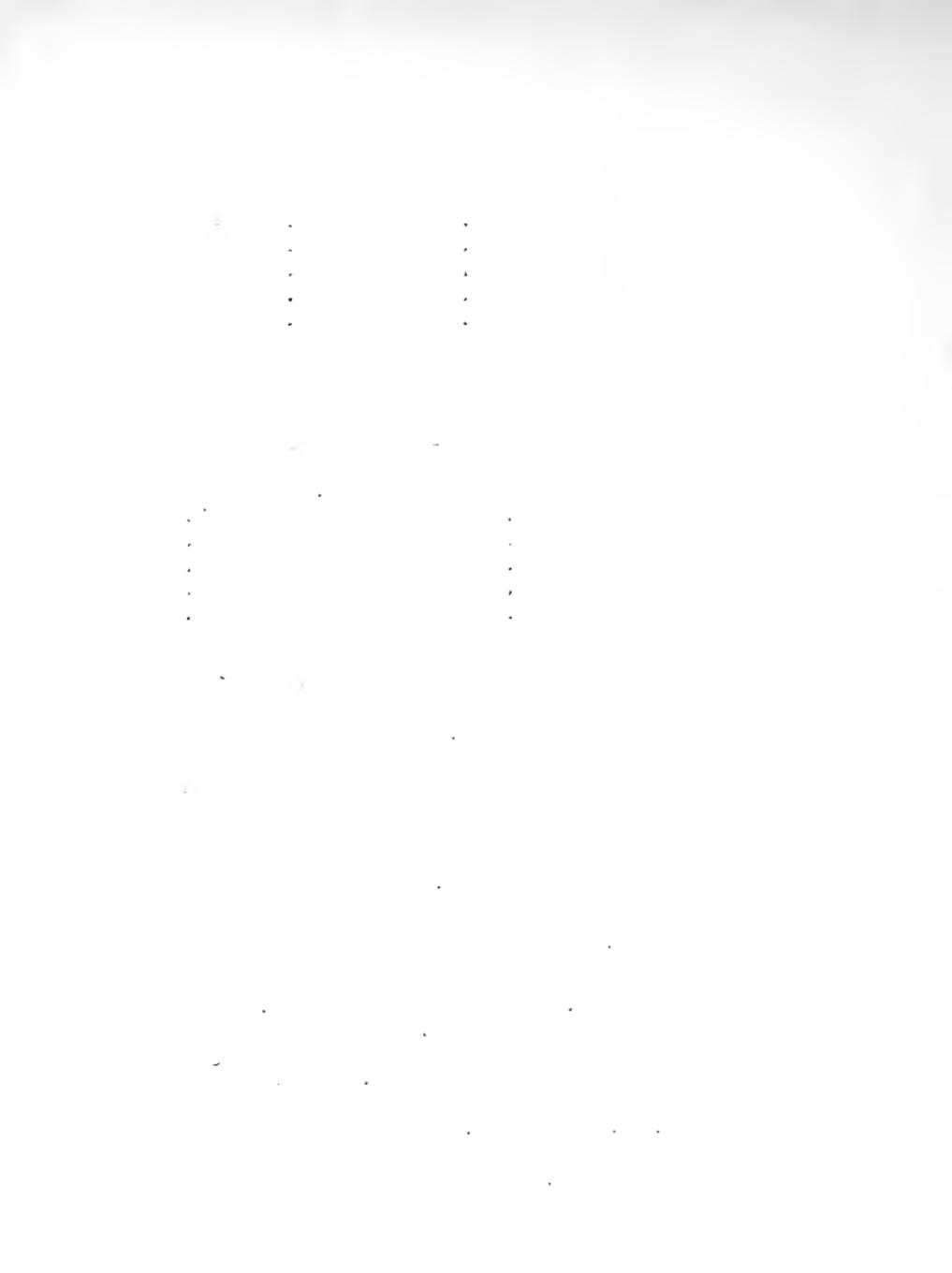
$S_1 = 3.26 \times t^2 = 3.26 \times .7^2 = 1.597.$

$S = \dfrac{D^1}{D} (S_1 - D) + D.$ $D = 14.625"$ $D_1 = 14.625"$

$S = 1 (1.597 - 1.22) + 1.22 = 1.597 \times 12 = 19.17"$

$A_1 = A \dfrac{(22.5 - 19.17}{22.5} = .148\ A$

Angle	Constant	A	A1
0°	.148	0	- 0
30°	"	- 9.94	- 1.47
60°	"	-17.21	- 2.55
90°	"	-19.85	- 2.94
120°	"	-17.21	- 2.55
150°	"	- 9.94	- 1.47
180°	"	- 0	0

$A_2 = A \times \dfrac{S}{L} = \dfrac{19.17}{22.5} = .853\ A.$

Angle	Constant	A	A2
0°	.853	0	- 0
30°	"	- 9.94	- 8.47
60°	"	-17.21	-14.66
90°	"	-19.85	-16.91
120°	"	-17.21	-14.66
150°	"	- 9.94	- 8.47
180°	"	- 0	0

CENTRIFUGAL FORCE OF THE CONNECTING ROD

The centrifugal force "C" of the connecting rod acts at the center of gravity, is always
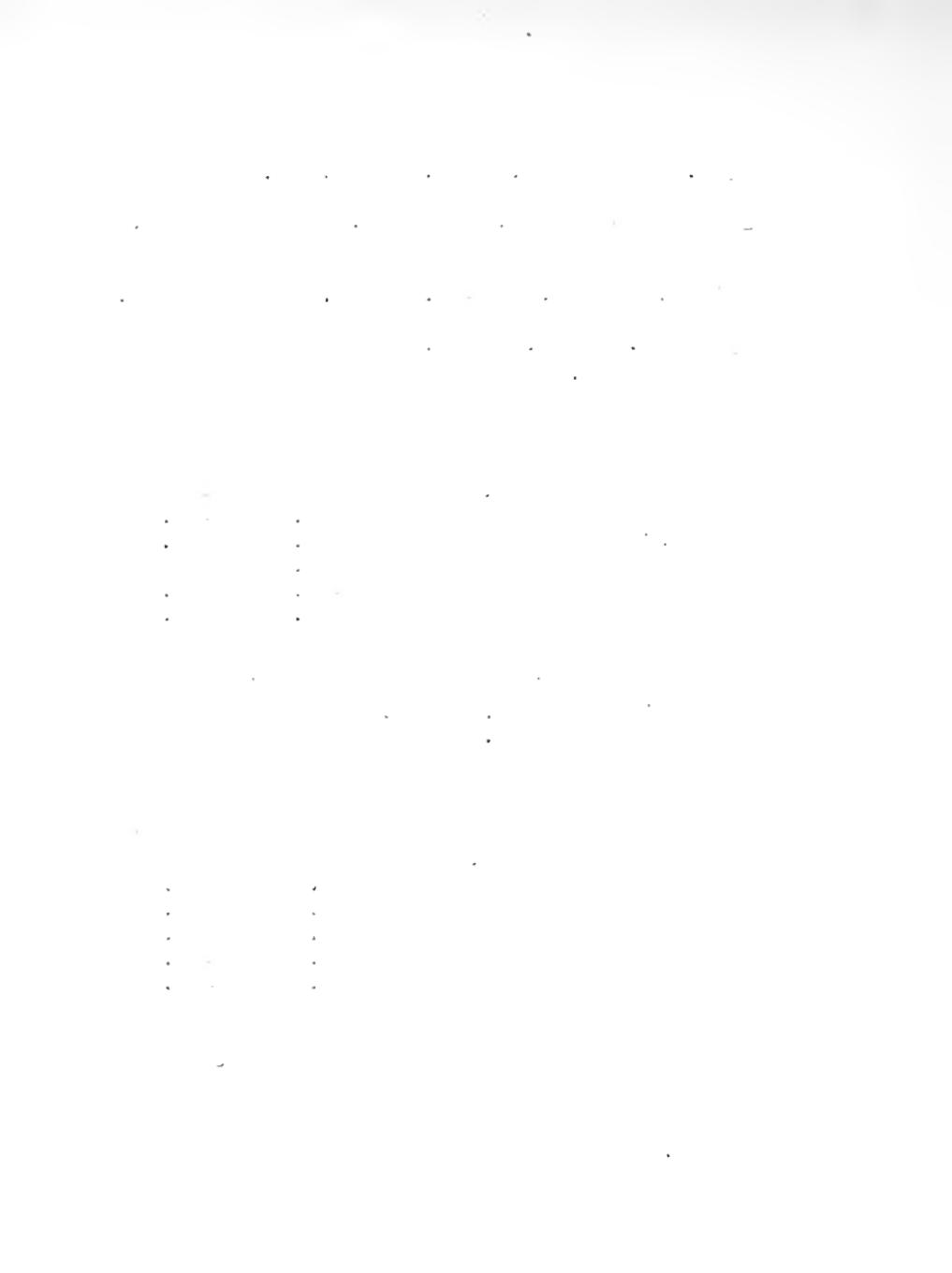
radial with the rod, and is outward in direction.
"C" can be determined from the equation

$$C = M_c \; W^2 \; Z_c \; (\frac{r}{L} \; \cos.\theta \;)^2.$$

$$M_c Z_o W^2 = 4^{\;2} \; x \; \frac{.953 \; x \; 56.25 \; x \; 14.625}{12 \; x \; 25.96} = 99.50$$

Angle	$\cos.\theta$	$1/5 \; \cos.\theta^2$
0°	1	.04
30	.866	.0299
60	.5	.01
90	0	0
120	.5	- .01
150	.866	- .0299
180	1	- .04

Angle	$1/5 \; \cos.\theta^2$	Constant	C
0°	.04	- 99.50	+3.98
30	.0299	"	+2.97
60	.01	"	+ .99
90	0	"	0
120	-.01	"	+ .99
150	-.0299	"	+2.97
180	-.04	"	+3.98

WEIGHT OF THE CONNECTING ROD

The influence of the weight of the connect-
ing rod is slight and is often neglected.

$$G_1 = \frac{G \ (L - Z_c)}{L}$$

$$G_1 = \frac{22.5 - 14.625}{22.5} \times 30.5 = .412$$
$$25.965$$

$$G_2 = G \ (\frac{Z}{L})$$

$$G_2 = \frac{14.625 \times 30.5}{22.5 \times 25.96} = .764$$

Since all the necessary forces have been calculated the resultant effect of the explosion pressure and inertia forces on the rotative effort of the crank pin, and on the shaking forces at the center of the main bearing can be determined.

The forces at the wrist pin are S_1 F_1 T_1 A_1 and G_1 resulting in a pressure on the guides and a thrust along the rod. This thrust along the rod combines with forces T_2, A_2, G_2, and C_1 to produce (1) rotation of the crank shaft

(2) pressure on the main bearings

(3) An unbalanced force tending to shake the engine on its foundation.
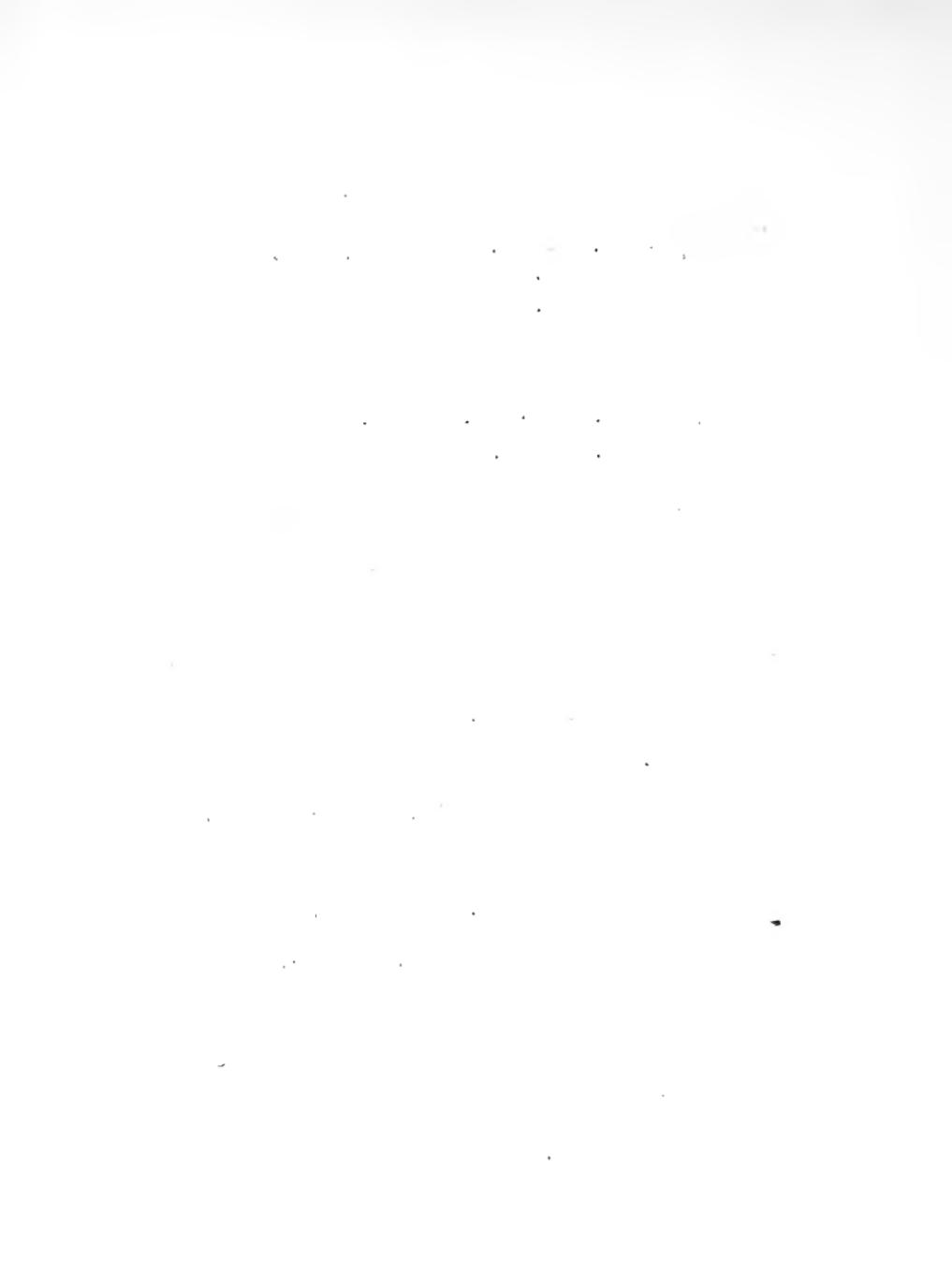
These various resultants have been determined
graphically in plates #1 and #2.

COUNTER BALANCE

The counter balance curve illustrates the
principle of reducing the effect of the shaking
forces by placing a counter balancing weight in
the crank disc of the engine opposite to the
crank pin. The centrifugal force of this weight
is uniform for a constant speed and is always
radial.

TURNING EFFORT

The force T E found on plate I in the force
polygons acts at right angles to the crank and
represents the turning effort in pounds per sq.
in. of piston area. While the piston makes a
stroke equal to twice the crank length, or 2 ,
the crank passes a distance   r. Hence, in
Fig. I, plate II, the turning effort at the va-
rious crank positions is laid off from the datum
line equal to 4   r. (2 revolutions per cycle
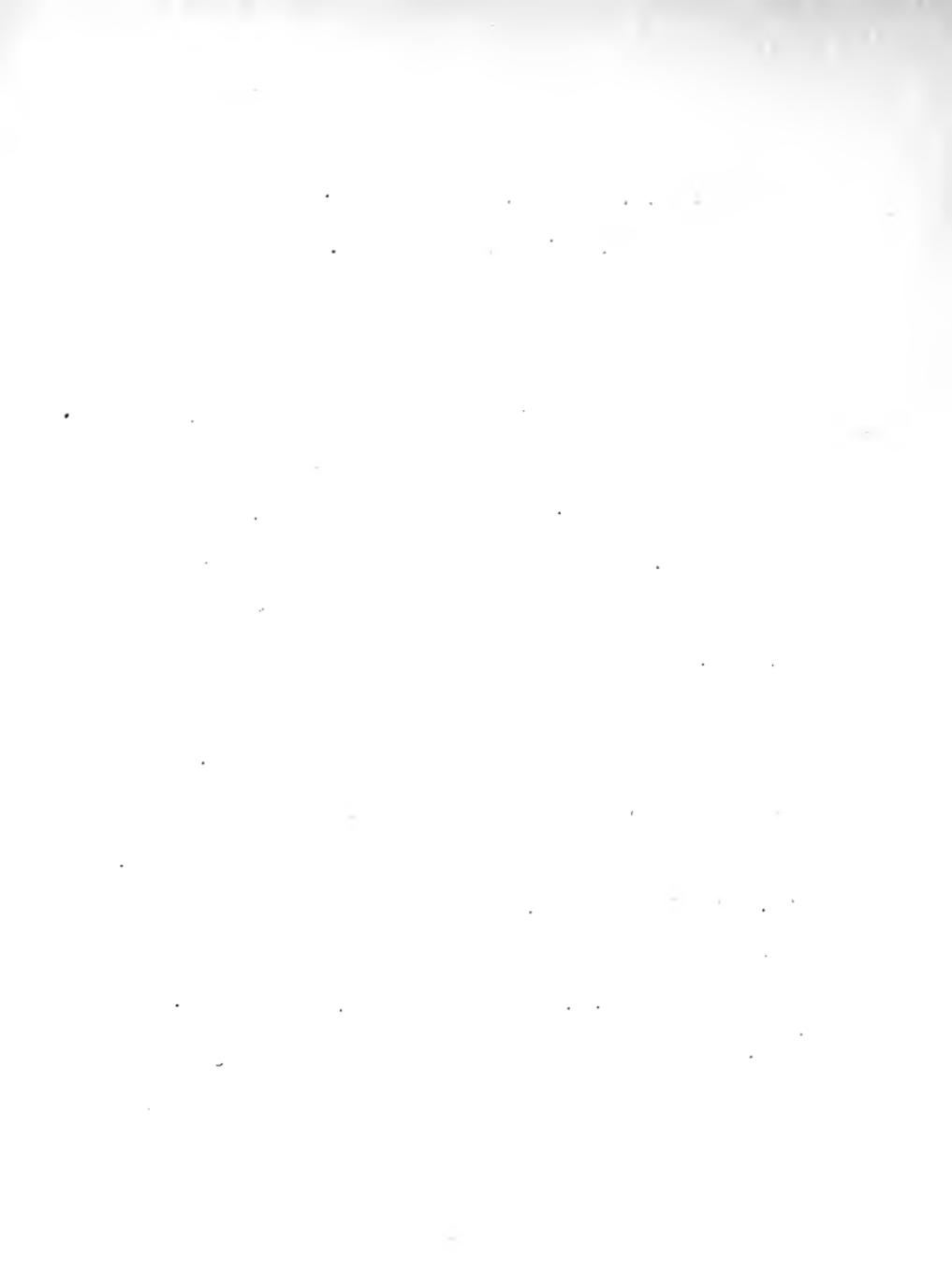
in a 4 cycle engine) at the point corresponding
to the crank pin position, in a circular path.
This process results in a curve and the area
between it and the datum line represents an
amount of work in foot pounds (equal to that of
the indicator card).

From the form of the turning effort curve
it follows that if the load is constant the speed
of the engine must fluctuate during each stroke.
The mean ordinate represents the uniform crank
effort which would do the same work per revolu-
tion as the actual effort. During part of the
stroke, the flywheel gains speed and stores up
kinetic energy which is restored during the
remainder of the stroke.

CALCULATION OF VELOCITY AND DISPLACEMENT CURVES

$F = S$ (Scale of spring) x P (ordinate in inches taken from mean ordinate of turning effort curve).

A = area of piston = 25.959 sq. inches.

$$d\,v = r\,d\,t \qquad = \frac{F\,R}{MK^2} \qquad K = \frac{15}{12}$$

$$r = \frac{(4.5)}{12} \qquad\qquad K^2 = \left(\frac{15}{12}\right)^2$$

$$d\,v = \frac{F\,R}{MK^2}\,r\,d\,t$$

Nt of 2 flywheels = 2 x 204 = 408

$$dt = \frac{60}{450 \times 36} = \frac{1}{270} = .00371.$$

$$dv = \frac{P \times 20 \times 25.959 \times 4.5^2 \times 12^2 \times 32.2 \times .00371}{12^2 \times 15.5^2 \times 408}$$

$$= dv = .0127\ P\ F\ T.$$

"DV" is then the velocity imparted to the flywheel, measured on the crank pin circle, during each interval by the force corresponding to P. The d V will then represent the total velocity imparted to the crank pin, measured from any given starting point.

The crank pin velocity curve is found by plotting d V's on a datum line, 4 r inches long,

divided into 36 equal parts. Each of the 36
equal parts correspond to those of the turning
effort curve. The resulting curve represents
the variation in velocity of the crank pin through
out 2 revolutions. By means of a planimeter the
mean ordinate of this curve was found, and drawn
giving a line of average velocity of crank pin.

After getting the crank pin velocity curve,
the crank pin displacement curve is found in a
similar manner.

$$d\ s = d\ v_s\ dt = .00371\ dv_s.$$

Since d s is the increment of space passed over
in each interval, it is evident that the d s =
total distance passed over by the crank pin during
any number of intervals. This summation was
plotted for every 10° and the resulting curve of
Fig. 3, plate II obtained. The mean ordinate
of this curve represents the displacement of the
crank pin, or the distance in feet through which
the crank would pass in unit time if its motion
were uniform.

TABLE I

TABULATION OF FORCES

0°	F	T	T_1	T_2	A	A_1	A_2	C	G_1	G_2
0	−29.60	− 36.06	−12.63	− 23.43	− 0	− 0	− 0	3.98	.412	.764
30	−24.05	− 29.40	−10.30	− 19.10	− 9.94	− 1.47	− 8.47	2.97	"	"
60	− 9.97	− 12.16	− 4.25	− 7.91	−17.21	− 2.55	−14.66	.99	"	"
90	+ 4.98	+ 6.072	+ 2.122	+ 3.95	−19.85	− 2.94	−16.91	0	"	"
120	+14.98	+ 18.29	+ 6.40	+ 11.89	−17.21	− 2.55	−14.66	.99	"	"
150	+19.13	+ 23.42	+ 8.22	+ 15.20	− 9.94	− 1.47	− 8.47	2.97	"	"
180	+19.95	+ 24.36	+ 8.53	+ 15.83	− 0	− 0	− 0	3.98	"	"
210	+19.13	+ 23.42	+ 8.22	+ 15.20	+ 9.94	+ 1.47	+ 8.47	2.97	"	"
240	+14.98	+ 18.29	+ 6.40	+ 11.89	+17.21	+ 2.55	+14.66	.99	"	"
270	+ 4.98	+ 6.072	+ 2.122	+ 3.95	+19.85	+ 2.94	+16.91	0	"	"
300	− 9.97	− 12.16	− 4.25	− 7.91	+17.21	+ 2.55	+14.66	.98	"	"
330	−24.05	− 29.40	−10.30	− 19.10	+ 9.94	+ 1.47	+ 8.47	2.97	"	"
360−720	−29.60	− 36.06	−12.63	− 23.43	+ 0	+ 0	+ 0	3.98	"	"
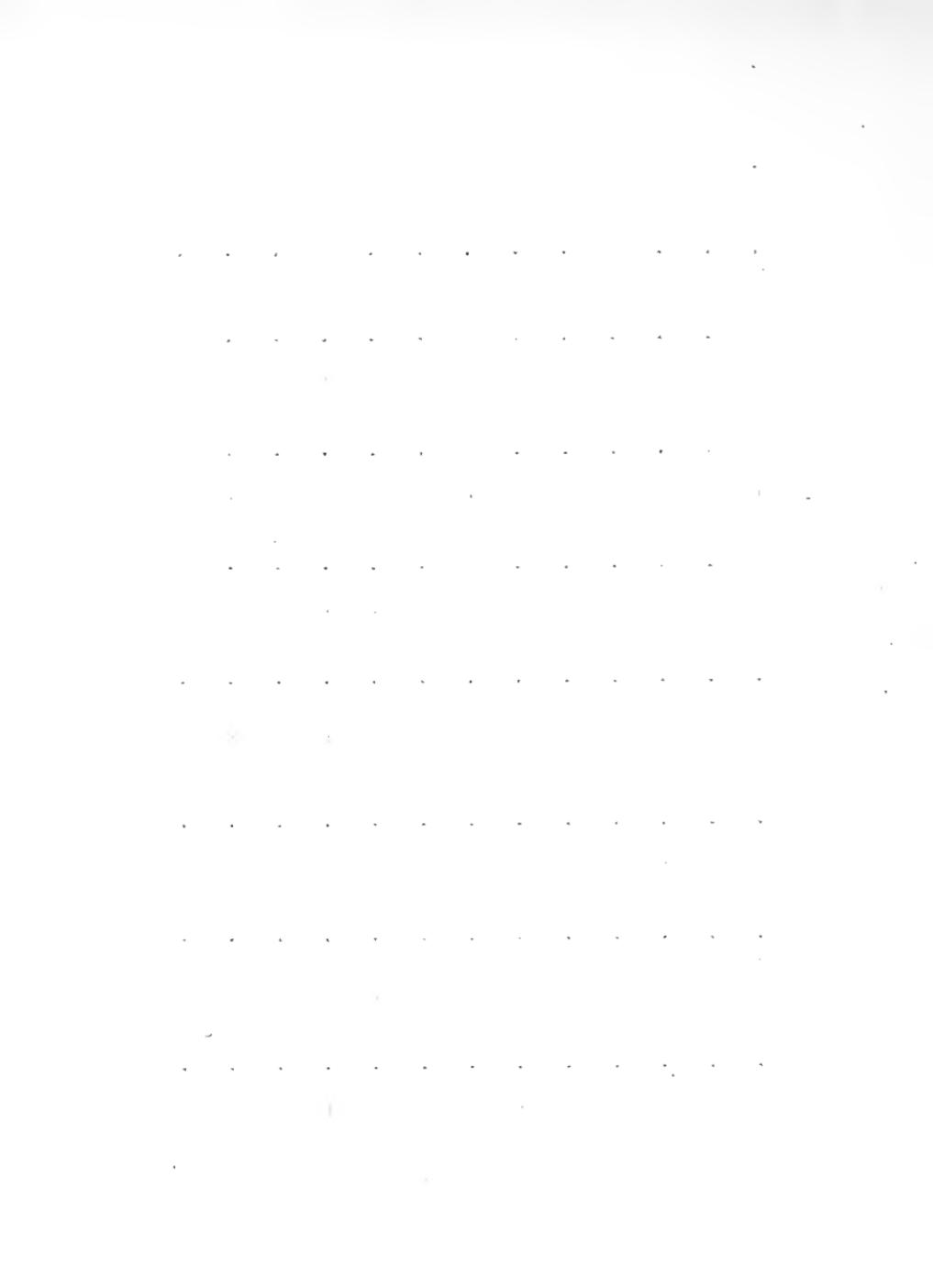
TABLE 2

VELOCITY

N⁰	θ°	P"	Px.0127 or dv = ft.	dv ft.
1	5	.04	.0005	.0005
2	15	3.00	.0381	.0386
3	25	7.45	.0945	.1331
4	35	7.33	.0930	.2261
5	45	6.57	.0834	.3095
6	55	5.50	.0698	.3793
7	65	4.44	.0564	.4357
8	75	3.70	.0470	.4827
9	85	3.23	.0410	.5237
10	95	2.95	.0374	.5611
11	105	2.75	.0348	.5959
12	115	2.48	.0314	.6273
13	125	2.17	.0276	.6549
14	135	1.83	.0232	.6781
15	145	1.32	.0162	.6943
16	155	.53	.0067	.7010
17	165	0	0	.7010
18	175	− .39	−.0049	.6961
19	185	− .68	−.0085	.6876
20	195	− .9	−.0115	.6761
21	205	− 1.12	−.0142	.6619
22	215	− 1.26	−.0160	.6459
23	225	− 1.38	−.0175	.6284
24	235	− 1.46	−.0185	.6099
25	245	− 1.46	−.0185	.5914
26	255	− 1.34	−.0169	.5755
27	265	− 1.14	−.0145	.5610
28	275	− .9	−.0114	.5496
29	285	− .53	−.0067	.5429
30	295	− .08	−.00111	.5418
31	305	+ .22	−.0028	.5446
32	315	+ .35	−.00445	.5480
33	325	+ .40	−.0057	.5537
34	335	+ .35	−.00445	.5581
35	345	+ .2	−.00254	.5606
36	355	− .18	−.00228	.5583
37	365	− .83	−.0105	.5478
38	375	− 1.23	−.0156	.5322
39	385	− 1.44	−.0183	.5139
40	395	− 1.53	−.0194	.4945
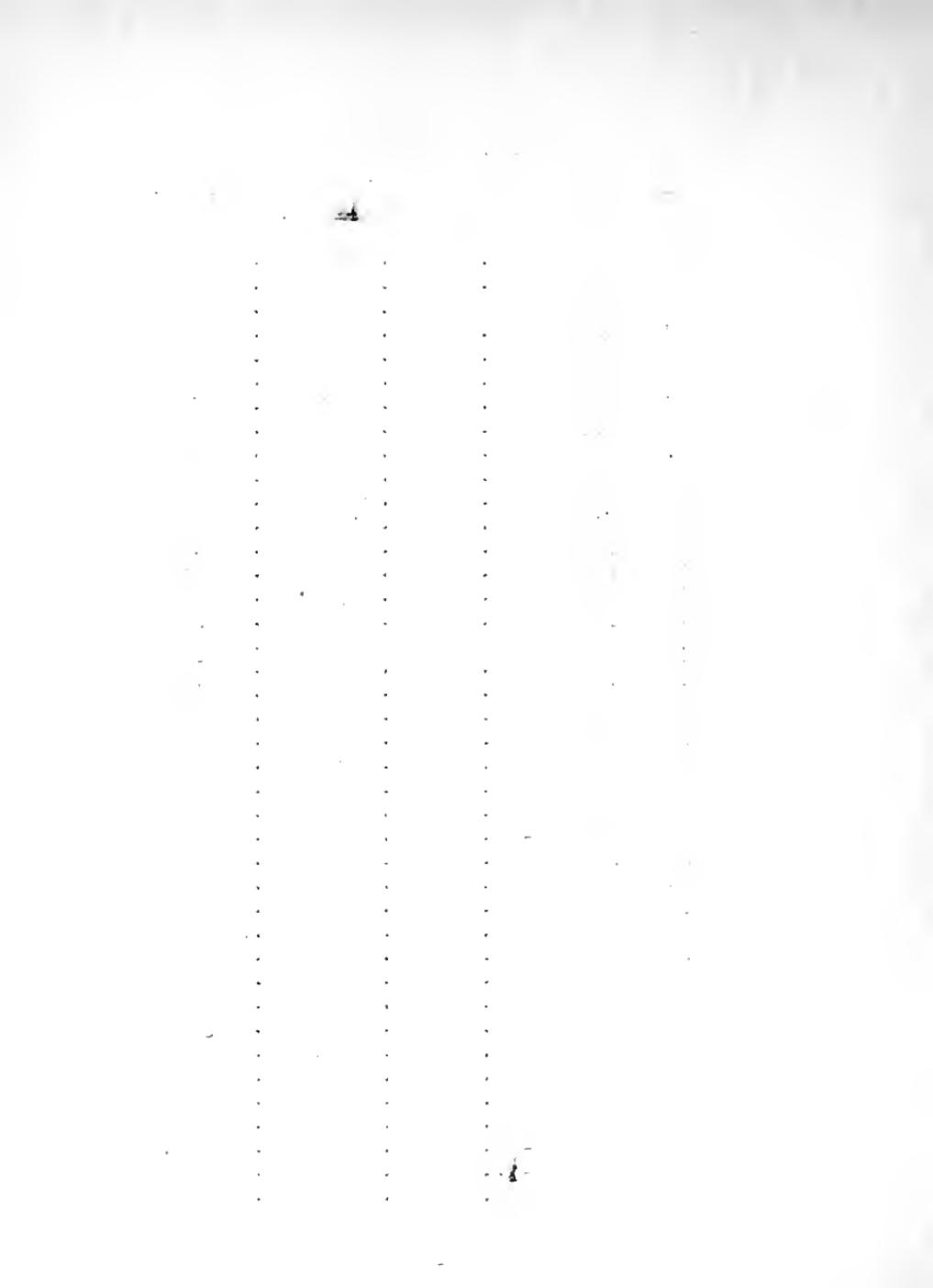
TABLE 2 (continued)

VELOCITY

N⁰	θ°	P"	Px.0127 or dv = ft.	dv = ft.
41	405	- 1.53	- .0194	.4751
42	415	- 1.44	- .0183	.4568
43	425	- 1.34	- .0170	.4398
44	435	- 1.24	- .0157	.4241
45	445	- 1.18	- .0156	.4091
46	455	- 1.06	- .0135	.3956
47	465	- .19	- .0024	.3932
48	475	+ .23	- .0029	.3961
49	485	.28	- .0036	.3997
50	495	.20	- .0025	.4022
51	505	.08	- .0011	.4033
52	515	- .07	- .0009	.4024
53	525	- . 2	- .00254	.3999
54	535	- .38	- .00482	.3951
55	545	- .62	- .0078	.3874
56	555	- .90	- .0114	.3759
57	565	- 1.25	- .0158	.3601
58	575	- 1.42	- .0180	.3421
59	585	- 1.47	- .0185	.3236
60	595	- 1.46	- .0185	.3051
61	605	- 1.42	- .0179	.2872
62	615	- 1.42	- .0179	.2693
63	625	- 1.46	- .0185	.2508
64	635	- 1.46	- .0185	.2323
65	645	- 1.42	- .0179	.2144
66	655	- 1.32	- .0165	.1979
67	665	- 1.35	- .0169	.1810
68	675	- 3.40	- .0425	.1385
69	685	- 4.40	- .0550	.0835
70	695	- 4.29	- .0535	.0300
71	705	- 3.07	- .0380	-.0080
72	715	- 1.29	- .0161	-.024
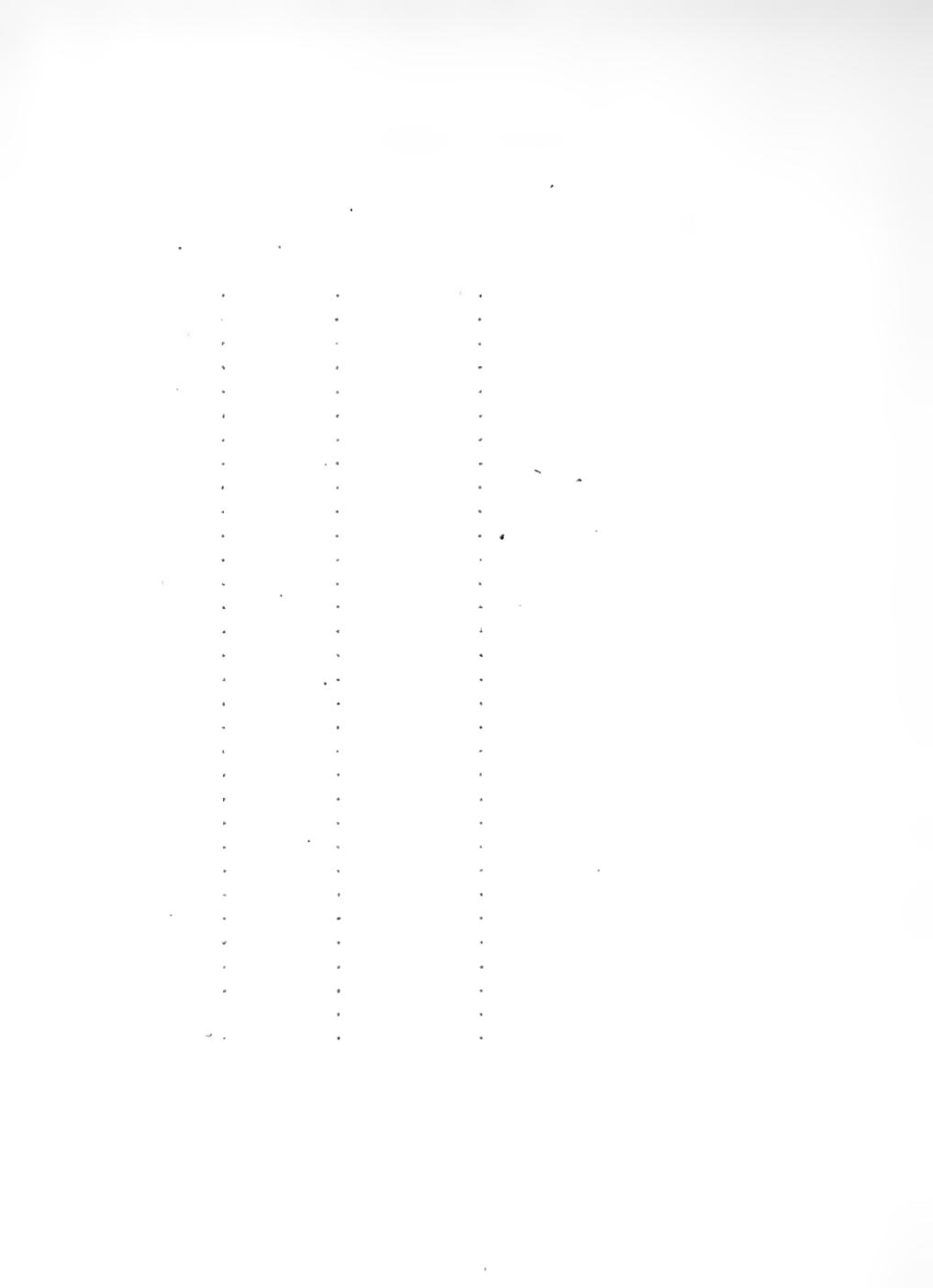
DISPLACEMENT TABLE

N⍛	θ	P"	Px.0037 or dvₛ= ft.	ds = ft.
1	10	- 4.02	- .0148	- .0148
2	20	- 3.40	- .0126	- .0274
3	30	- 2.35	- .0078	- .03526
4	40	- 1.5	- .0056	- .04082
5	50	- .64	- .0024	- .04319
6	60	- .08	- .00029	- .04348
7	70	+ .43	+ .00159	- .04189
8	80	+ .80	+ .00296	- .03893
9	90	+ 1.21	+ .0045	- .03443
10	100	+ 1.55	+ .00575	- .02868
11	110	+ 1.88	+ .00696	- .02172
12	120	+ 2.22	+ .00816	- .01356
13	130	+ 2.48	+ .0092	- .0043
14	140	+ 2.65	+ .00985	+ .0054
15	150	+ 2.77	+ .0103	+ .01519
16	160	+ 2.77	+ .0103	+ .0261
17	170	+ 2.70	+ .0100	+ .0361
18	180	+ 2.60	+ .00965	+ .0451
19	190	+ 2.56	+ .0095	+ .0552
20	200	+ 2.30	+ .0085	+ .0636
21	210	+ 2.16	+ .008	+ .0716
22	220	+ 1.97	+ .0073	+ .0789
23	230	+ 1.79	+ .0066	+ .0855
24	240	+ 1.62	+ .0060	+ .0915
25	250	+ 1.47	+ .0055	+ .0970
26	260	+ 1.33	+ .00493	+ .1019
27	270	1.24	.0046	.1065
28	280	1.20	.0045	.1109
29	290	1.24	.0046	.1155
30	300	1.27	.0047	.1202
31	310	1.32	.0049	.1252
32	320	1.36	.0050	.1302
33	330	1.39	.0051	.1353
34	340	1.37	.0051	.1404
35	350	1.30	.0048	.1452
36	360	1.19	.0044	.1496
37	370	1.02	.00379	.1534
38	380	.84	.00312	.1565
39	390	.63	.00234	.1588
40	400	.42	.00156	.1604
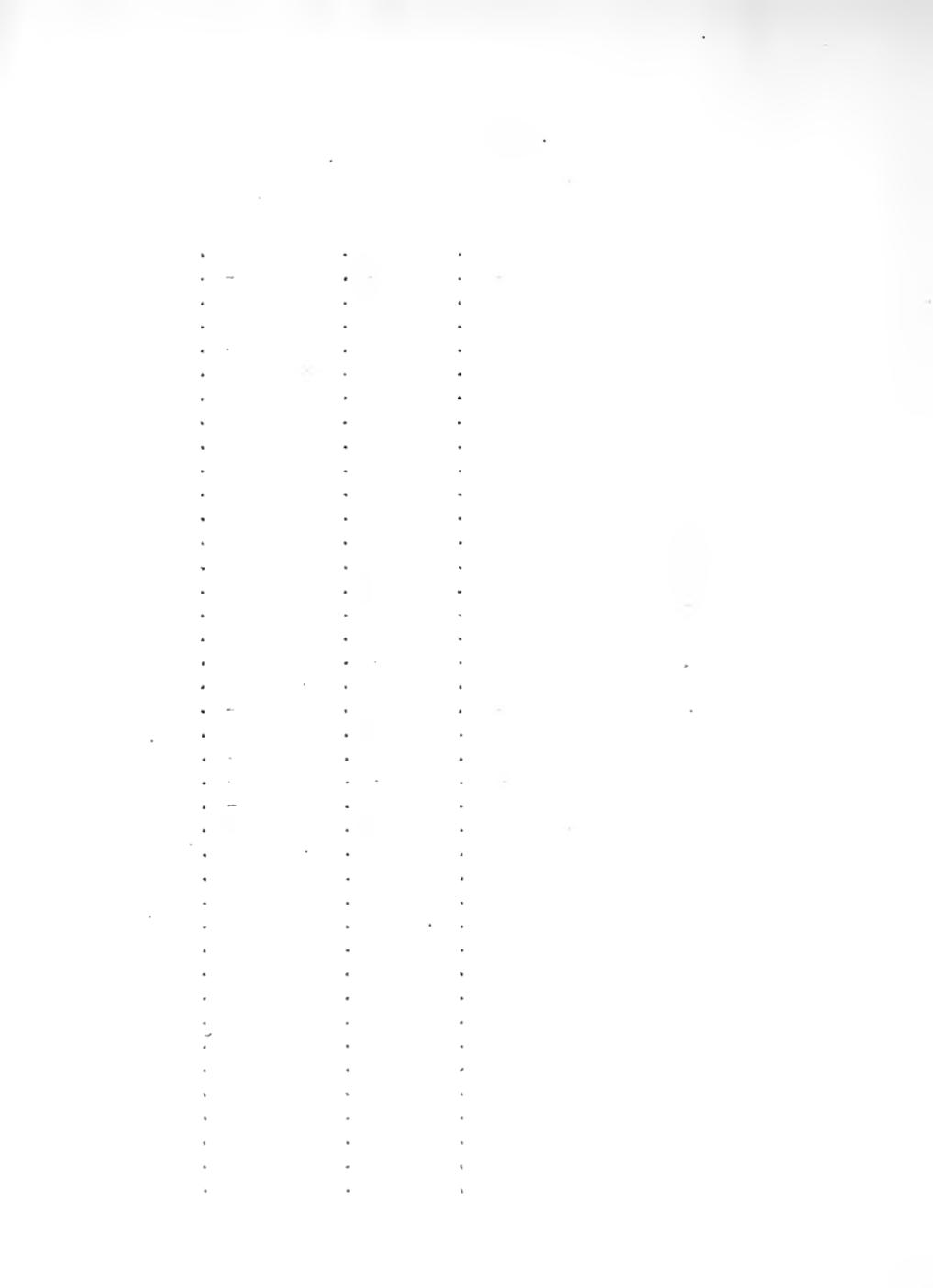
No.	θ	P"	ds = ft. or Px.0037	ds = ft.
41	410	.26	.00096	.1614
42	420	+ .10	+ .000371	.1617
43	430	- .06	- .00022	.16150
44	440	- . 2	- .00074	.1608
45	450	- .28	- .00104	.1587
46	460	- .30	- .00111	.1576
47	470	- .28	- .00104	.1557
48	480	- .22	- .00082	.1548
49	490	- .20	- .00074	.1541
50	500	- .21	- .00077	.1533
51	510	- .24	- .00089	.1524
52	520	- .27	- .001	.1514
53	530	- .32	- .00118	.1502
54	540	- .42	- .00156	.1487
55	550	- .54	- .0020	.1467
56	560	- .71	- .00263	.1440
57	570	- . 9	- .00334	.1407
58	580	- 1.09	- .00404	.13669
59	590	- 1.26	- .00467	.1320
60	600	- 1.45	- .00537	.12665
61	610	- 1.65	- .0061	.12055
62	620	- 1.84	- .0068	.11373
63	630	- 2.04	- .00756	.10617
64	640	- 2.21	- .0082	.09697
65	650	- 2.39	- .0088	.08810
66	660	- 2.65	- .0098	.07826
67	670	- 3.17	- .0117	.06656
68	680	- 3.72	- .0138	.05276
69	690	- 4.09	- .0152	.03756
70	700	- 4.20	- .0155	.0220
71	710	- 4.23	- .0157	.0063
72	720	- 4.23	- .0157	- .0094

PLATE 1

FORCE POLYGONS

A TYPICAL H.V.ID CARD

SCALE
500 : 1

AREA - 40.7 □"

PISTON DISPLACEMENT

EXHAUST STROKE

SHAKING FORCE CURVE
SCALE 4 : 1

COUNTER BALANCE CURVE
SCALE 50 : 1

COUNTER BALANCE CURVE
SCALE 50 : 1

COMPRESSION STROKE

PRESSURE ON GUIDES
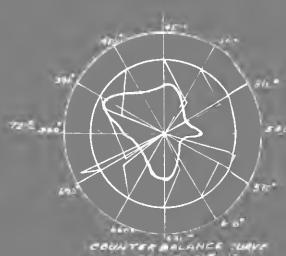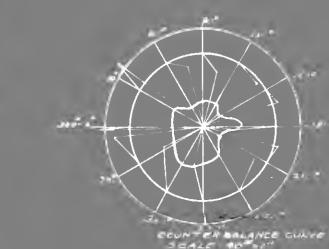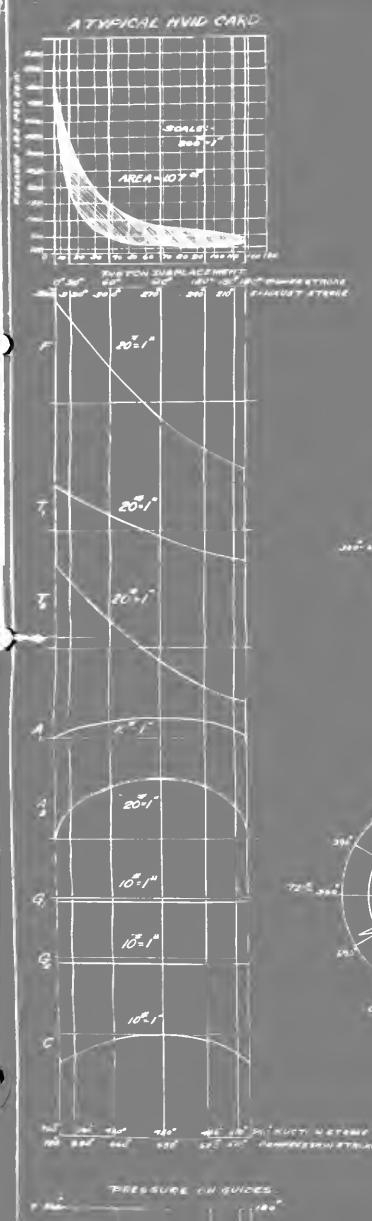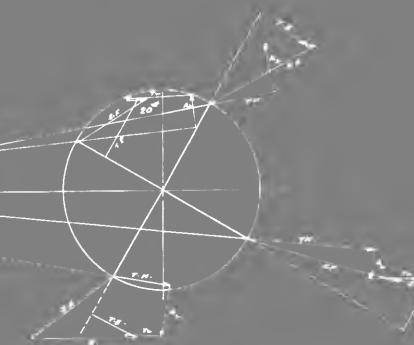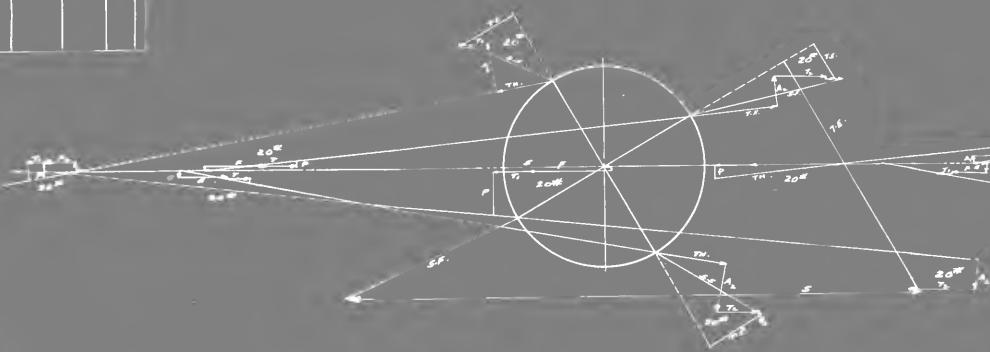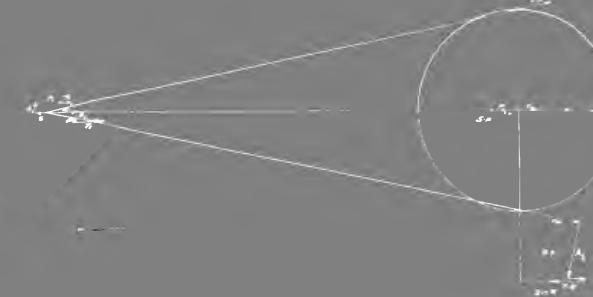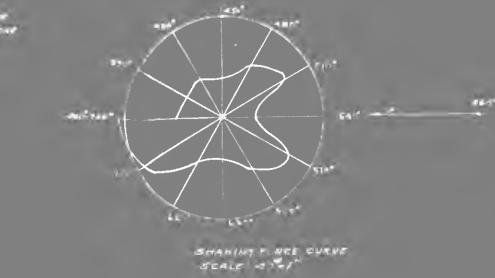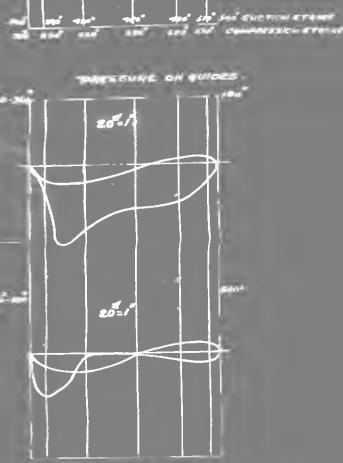
SHAKING FORCES

OF AN

8 HP HVID ENGINE

450 R.P.M.

JONES & SCHWACHTGEN THESIS

1920

DRAWN BY
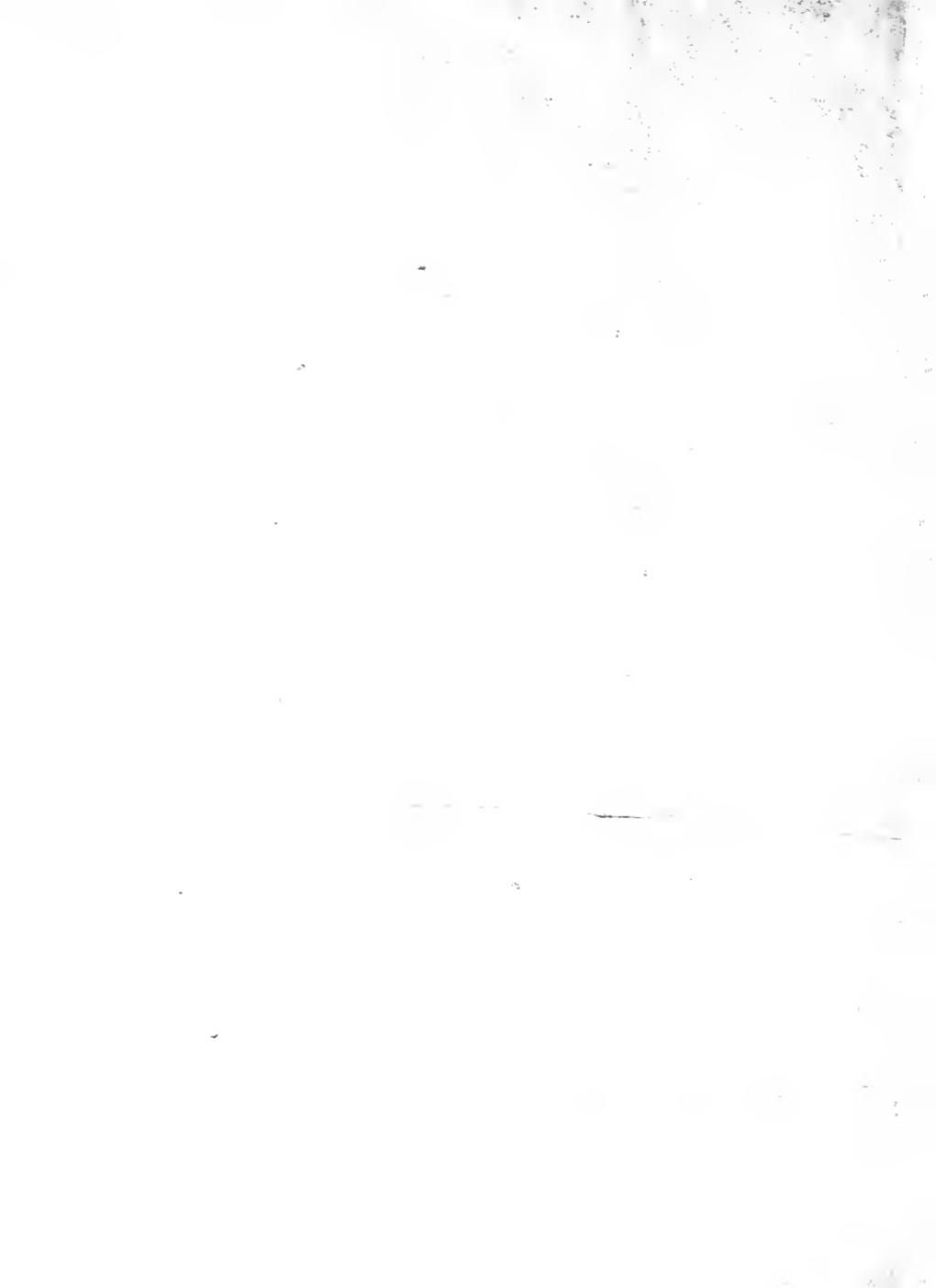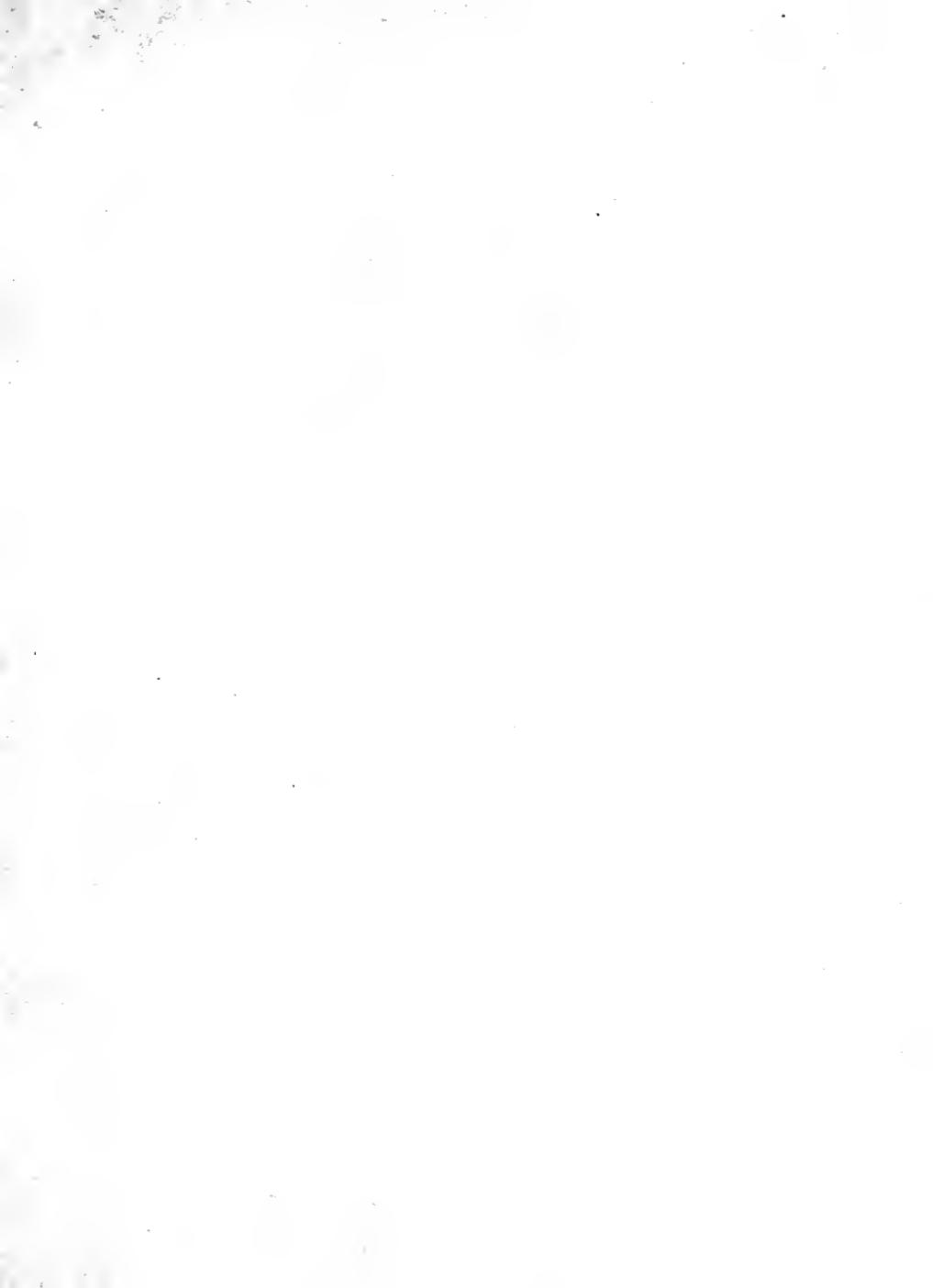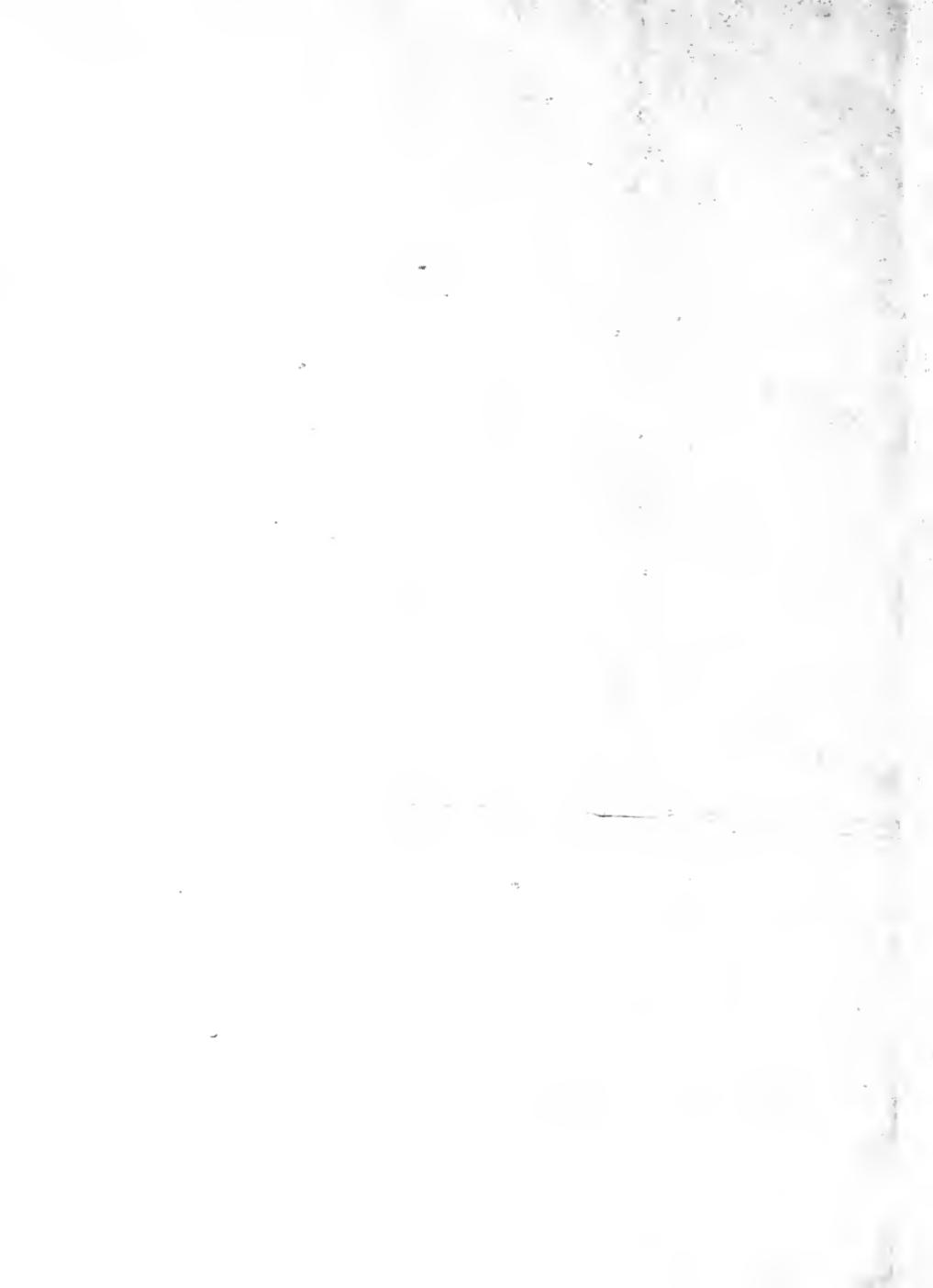
PLATE 2

FIG. 1

"TURNING EFFORT CURVE"
AREA = 10.6□"
LENGTH = 17.75"

FIG. #2

VELOCITY CURVE
AREA = 7.14□"
LENGTH = 17.75"

FIG. #3

DISPLACEMENT CURVE
AREA = 45.83□"
LENGTH = 17.75"

SHAKING FORCES
8 H.P. HVID ENGINE

450 R.P.M.
JONES & SCHWACHTGEN THESIS
1920

$\frac{3}{8}''$

$\frac{3}{4}$

$\frac{1}{8}''$

$3\frac{1}{4}''R$

6¾"

3"
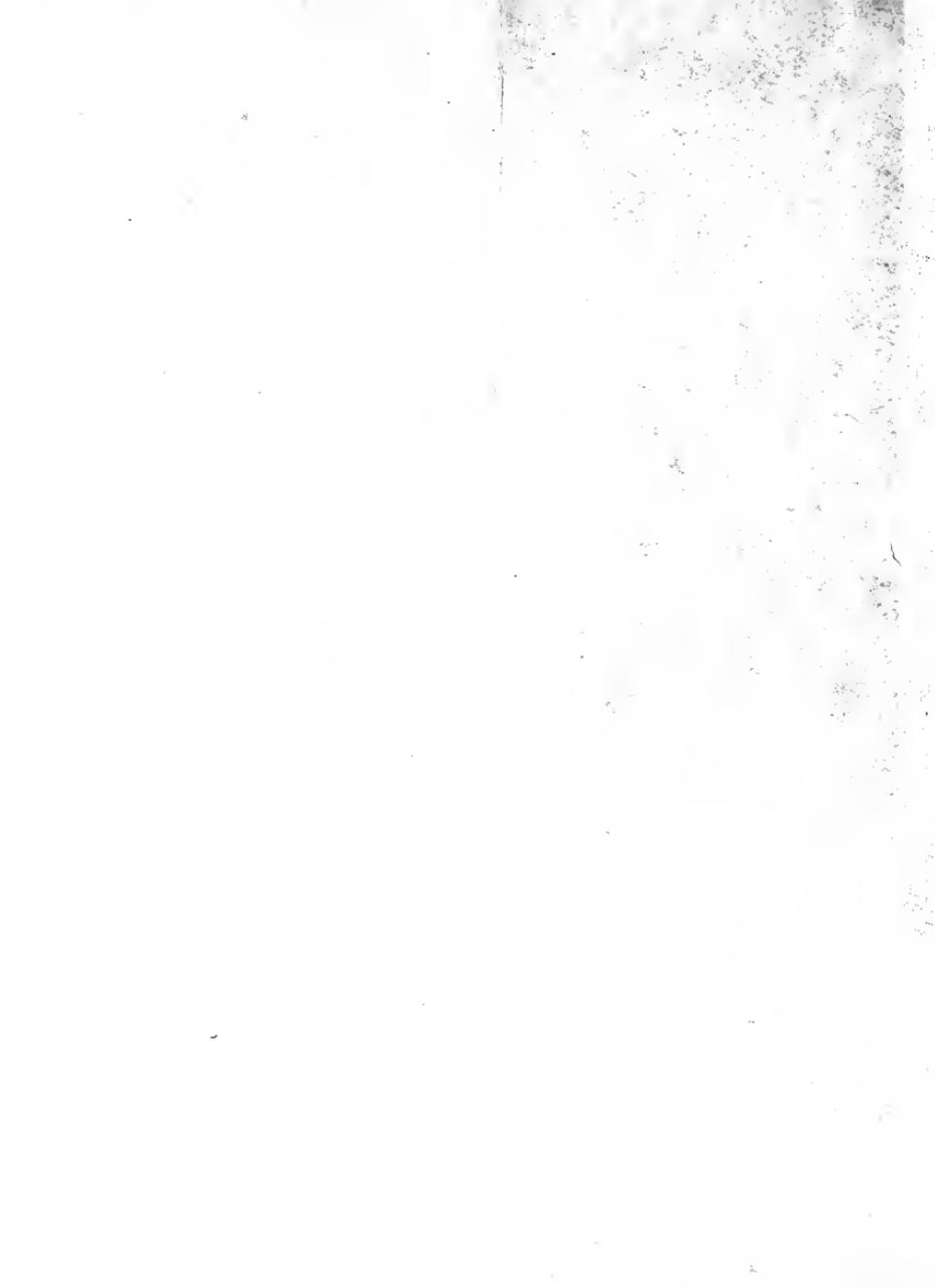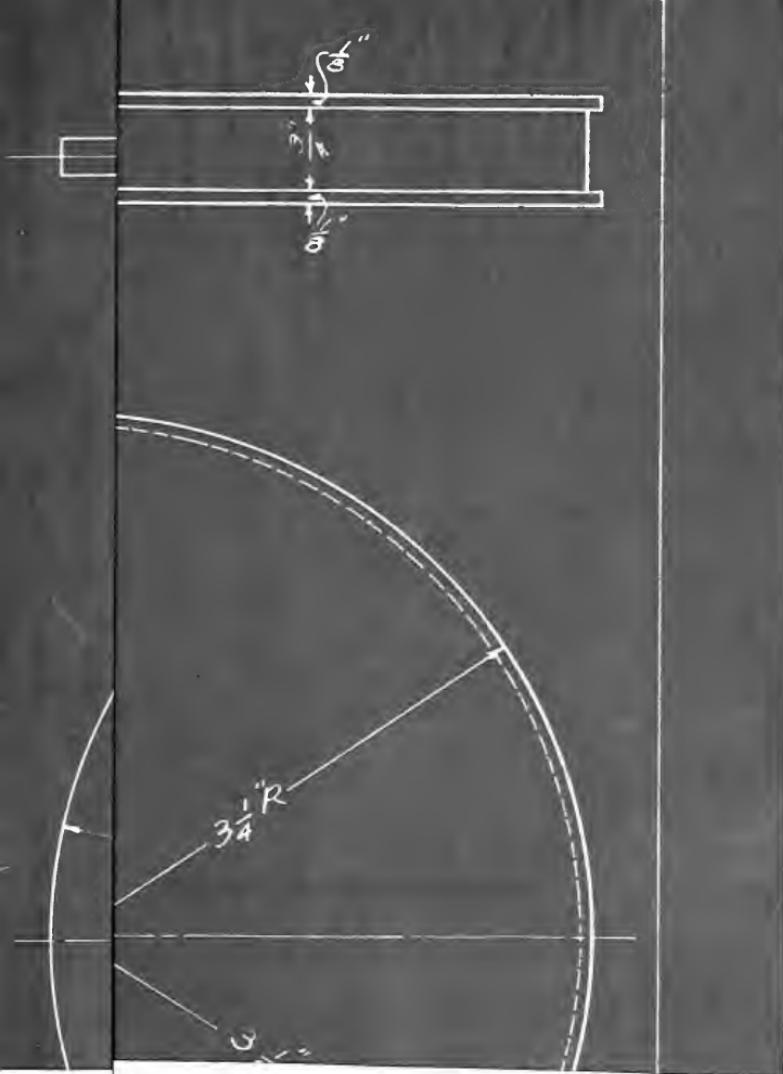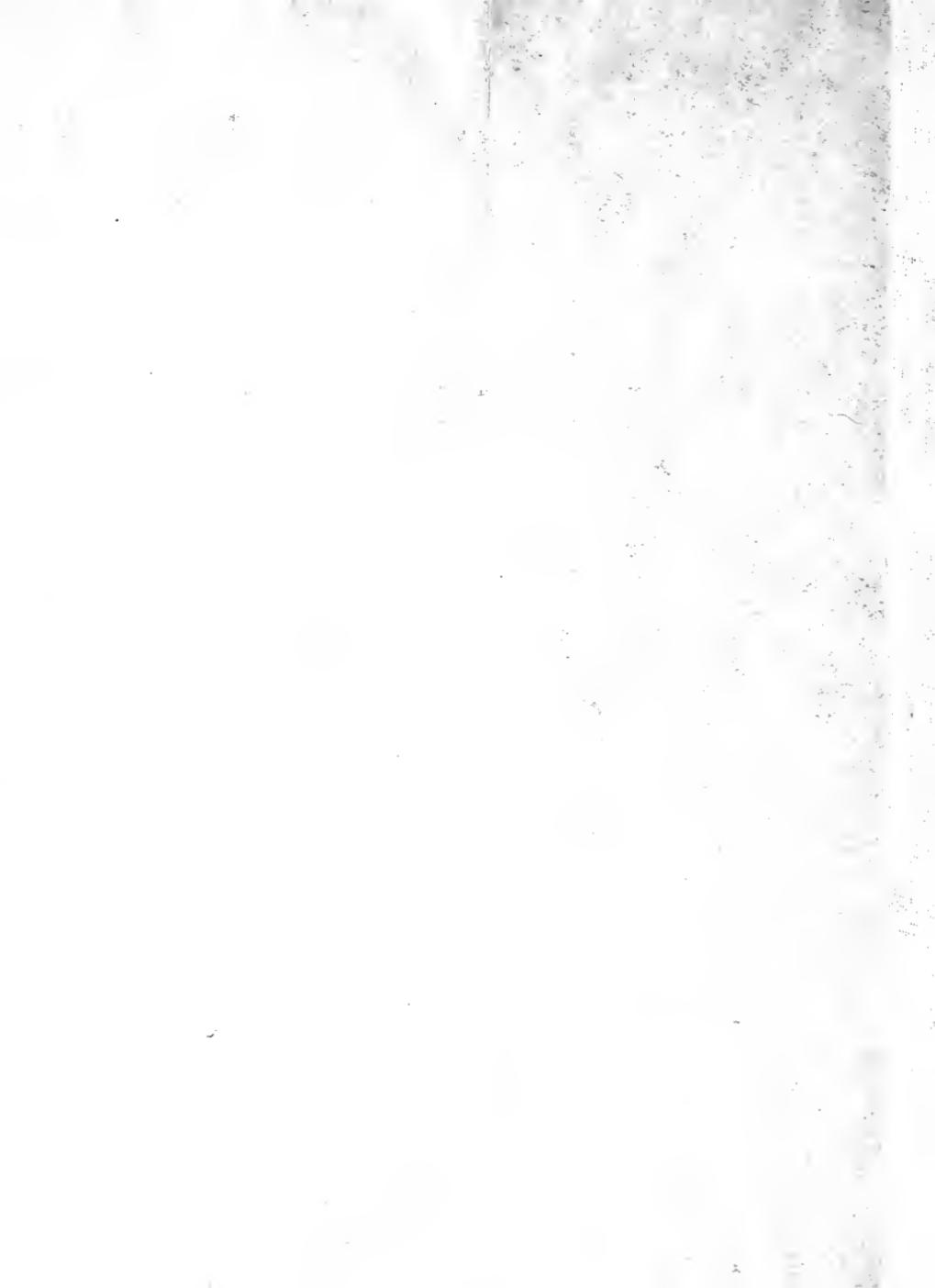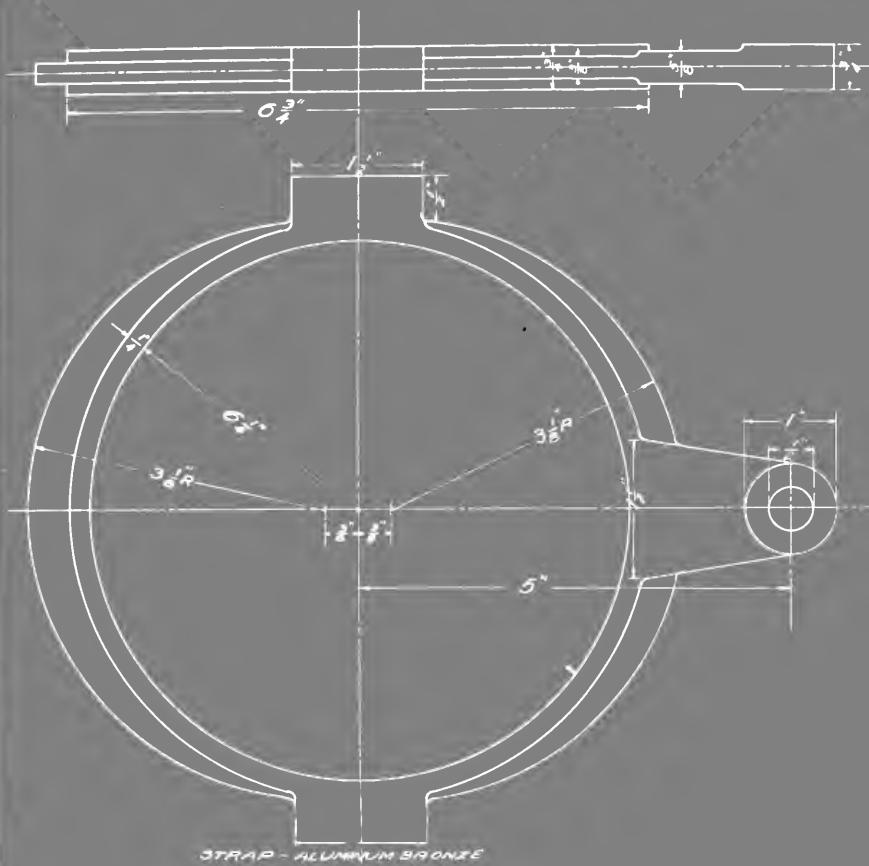
STRAP - ALUMINUM BRONZE

SHEAVE - C.I.

DESIGN OF
ECCENTRIC
FOR REDUCING MOTION ON
MV10 ENGINE
MAR 20. 20   THESIS
L.E.JONES
A.I.T.

6

DETAIL OF ANGLE REINFORCEMENT.

FIXED TRAM ROD - $\frac{5}{8}$ C.R.S. THREADED DOWN.

TO BE TWISTED HOT.

THIS DIMENSION SCALED TO - ON - $31\frac{1}{2}$

$22\frac{1}{2}$

DETAIL OF EXISTING.

CHANGES
NOTE:- USE $\cdot36 \times \frac{1}{2}$ CARR'G BOLTS FOR BRAKE BLOCKS AND DRILL

ASSEMBLY OF BRAKE

1. BRAKE

DETAIL OF ANGLE REINFORCEMENT.

BILL OF MATERIAL

No	PART	M'L	No REQ'D	REMARKS
1	BRAKE BLOCK	Wood	12	$8\frac{1}{2}" \times 8" \times 1\frac{1}{2}"$
2	BRAKE BANDS	W.I.	3	$1\frac{1}{2}" \times 11\frac{1}{2}" \times \frac{1}{4}"$
3	SPRINGS	St.I.	2	$1" \times 1\frac{1}{2}"$ OR $\frac{1}{2}$ PITCH
4	BRAKE ARM STAYS	W.I.	2	$1"$
5	BRAKE ARMS	W.I.	6	$1\frac{1}{2}" \times 26\frac{1}{2}" \times \frac{1}{4}"$
6	$\frac{3}{4}"$ STAYS	W.I.	6	
7	$\frac{3}{4}"$ BOLT & NUT	W.I.	1	$3"$ LONG
8	$\frac{3}{4}"$ " "	W.I.	36	$3"$ LONG
9	$\frac{3}{4}"$ WASHERS	W.I.	6	
10	$\frac{3}{4}"$ U-BOLT	W.I.	1	$10"$ LONG, THREADED
11	ANGLE SUPPORT	W.I.	2	$1\frac{1}{2}" \times 7\frac{1}{2}" \times \frac{1}{4}"$
12	" REINFORCEMENT	W.I.	2	$1\frac{1}{2}" \times 7\frac{1}{2}" \times \frac{1}{4}"$
13	WASHERS	W.I.	72	

12 BLOCKS AS SHOWN

TO BE COUNTERSUNK AS SHOWN

NOTE: ALL HOLES TO BE DRILLED AFTER BANDS ARE SHAPED.

DETAIL OF BRAKE BANDS WITH BLOCKS.

SCALE: 3" = 1 FT.

DETAILS OF
24" WOODEN BRAKE
DESIGNED FOR JONES D. SCHWACHTGEN THESIS
ARMOUR INSTITUTE OF TECHNOLOGY
1920
DRAWN BY: CHECKED BY: